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DESCRIPTION AND TEST OF A QUIMBY SCREW PUMP.

BY LIEUT.-COMDR. W. R. PURNELL, U. S. N.

Tests of the Quimby Screw Pump to determine its suitability for use in the Naval Service as a high pressure fuel oil service pump were conducted at the Fuel Oil Testing Plant, Navy Yard, Philadelphia, Pa.

The pump tested was supplied by the Wm. Quimby, Incorporated, 209 Parkhurst Street, Newark, N. J., the sole manufacturer. It was of standard make, especially designed for high pressure work and listed as No. 3½ High Pressure Fuel Oil Pump. It was directly connected to a 15-horsepower 800-1,450 revolutions per minute, 230-volt, D.C. motor.

The general assembly of the pump with motor, or turbine, direct drive is shown in Figure 1 and Figure 2, and the details of construction in Figures 3 and 4.

The four screws that act as pistons in propelling the oil through the pump chamber are mounted in pairs on parallel

shafts, and are so arranged that in each pair the thread of the screw projects to the bottom of the space between the threads of the opposite screws. The screw threads have flat faces and peculiarly undercut sides; the width of the face and base of the thread being one-half the pitch. Sufficient clearance is left between the screws and the cylinder wall, and between the faces of the intermeshing threads to allow a close running fit without actual contact.

The suction connection shown at "S," Figure 1, can be made on either side of the pump, and opens into a suction chamber underneath the pump cylinder. The oil passes through this chamber to the two ends of the cylinder where it is trapped

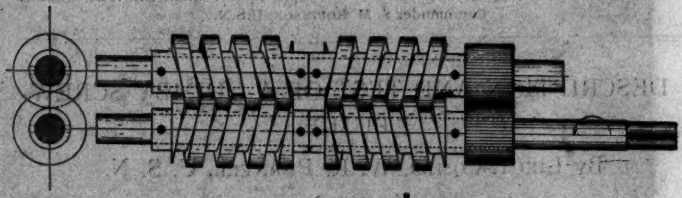
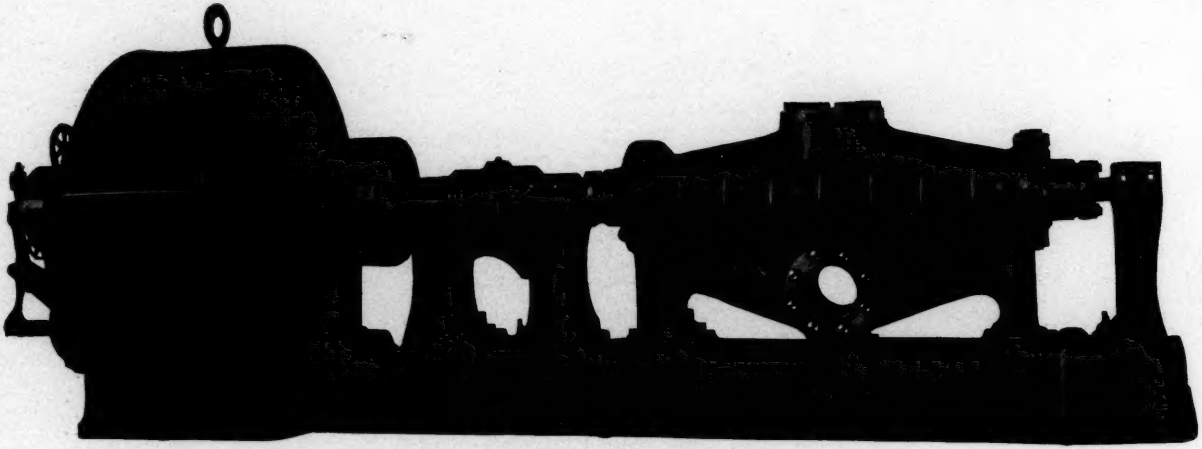


FIG. 3.

by the screws and forced from the two ends towards the center discharge chamber by the action of the two pairs of intermeshing threads. Each revolution of the pump delivers positively the contents of the space between one convolution of each of the threads less only the small amount due to clearance between the screws, and between the screws and the cylinder. The discharge connection is shown at "D" in the top of the cylinder. The discharge connection causes the back pressure of the discharge to be delivered against the middle of the cylinder and eliminates end thrust of the screws *in* their bearings, due to the fact that the endwise pressure upon the screws in one direction is exactly counter-balanced by a like pressure in the opposite direction.

FIG. 1.



85 G. P. M. 300 Pounds Pressure Fuel Oil Pump with 40 H. P. 1200 R. P. M. Terry Turbine.
Viscosity of Oil 300 Degrees Engler.

Fig. 2.

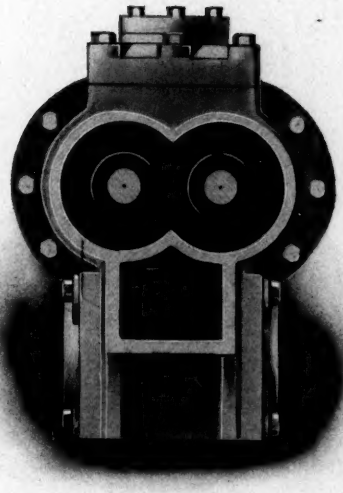


FIG. 4.

The power to drive the pump is applied directly to one of the shafts and the second shaft is driven by means of a pair of gears shown at "G" in Figure 1.

The pump has no valves, no internal packing, and no small moving parts. The working parts are comprised of two moving pieces, the two shafts with the screws and the gears that are rigidly secured to them. The shafts pass through stuffing boxes in the cylinder head. As these boxes are on the suction side there is no tendency to blow out the packing. Sufficient internal lubrication is furnished by the oil being pumped. The pump cylinder is made of a tough grade of gun iron and the worms of hard gun iron. The bearings are of Navy bronze. The pumps are manufactured in different sizes and of varying construction depending upon the capacity and the discharge pressure. Capacities range from 2 to 8,000 gallons per minute and discharge pressure from 0 to 600 pounds per square inch gauge.

The pump as installed at the Testing Plant was fitted with a 2-inch suction pipe and a 1½-inch discharge line. It was directly connected to a 15-horsepower variable speed motor, the efficiency of which was taken as 86.9 per cent and considered constant.

The tests were conducted for capacity, efficiency and pressure. The suction was taken from a 450-gallon supply tank and the pump discharged through two oil meters in series and a throttling valve into two weighing tanks. A calibrated voltmeter and ammeter were used to indicate the power supplied to the motor.

The series of tests were made under varying speeds and pressures as shown by the curves. The discharge pressure variation was from 100 to 450 pounds per square inch gauge and the speed variation from 740 to 1,210 revolutions per minute. The first tests were conducted with ten threads on each screw. After the first series the screws were removed and the number of threads was reduced to eight.

| Run No. | Duration | Motor | | | | Pump | | | |

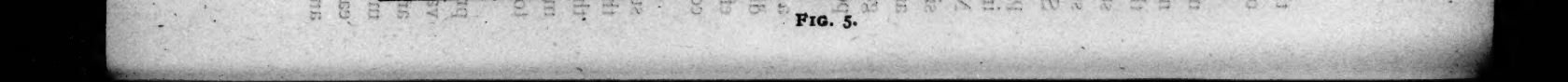


FIG. 5.

SUMMARY SHEET OF TEMPERATURE - CAPACITY DATA.

Run No	Duration	Motor				Speed of Motor and Pump	Pump												
		Voltmeter	Ammeter	Input	Output		Oil Temp.		Oil Press		Discharge by Engler Meter			Gal Discharge for 1000 R.P.M.	Theoretic Discharge	Slip	Input to pump	Output from pump	Eff.
							Suct.	Disch	Suct.	Disch	Total	Per Min	Corrected						
	Min	Volts	Amps.	K.W.	K.W.	R.P.M.	°F	°F	Lbs/Sq. in.		Gal.	G.P.M.	G.P.M.	G.P.M.	G.P.M.	%	H.P.	H.P.	%
22	20	214.2	27.0	5.78	5.02	819	56.6	56.7	1.0	250.0	324	16.20	25.38	25.28	36.04	28.2	6.78	3.99	56.0
24	16	214	26.0	5.55	4.82	891	74.8	74.8	1.0	250.0	357	22.19	23.87	23.00	36.56	34.6	6.46	3.48	53.7
26	24	217.0	25.0	5.43	4.72	848	87.0	87.1	1.3	250.0	508	13.25	22.76	21.66	37.31	38.5	6.33	3.35	52.9
28	20	219.8	25.3	5.66	4.83	859	101.0	100.8	1.3	250.0	447	22.46	22.17	20.79	37.53	40.7	6.47	3.23	47.7
30	8	221.7	25.0	5.51	4.86	862	108.7	107.0	1.0	250.0	174	21.75	21.48	19.74	37.75	43.1	6.51	3.13	48.1
32	20	218.8	25.3	5.39	4.69	819	110.2	107.5	1.0	250.0	394	19.70	19.46	17.01	36.04	46.0	6.21	2.84	45.7
34	20	211.2	27.0	5.70	4.95	828	120.3	121.0	1.0	250.0	384	19.20	18.96	16.32	36.19	48.0	6.64	2.76	41.6
														Oil Disch for 1000 R.P.M.					
23	16	207.0	36.6	7.65	6.65	1070	66.2	66.0	2.0	250.0	579	35.81	35.31	35.61	47.76	26.3	8.71	5.16	59.7
25	12	207.3	34.8	7.21	6.27	1075	81.0	81.0	3.0	250.0	417	34.93	33.71	34.04	48.13	29.6	8.40	4.94	58.8
27	12	216.5	34.0	7.14	6.20	1131	96.0	95.5	2.5	250.0	411	34.26	33.83	32.70	47.76	31.0	8.21	4.79	58.3
29	12	212.3	30.0	6.10	5.32	1142	105.0	104.8	2.5	250.0	408	34.00	33.58	32.35	50.25	33.7	7.80	4.87	62.4
31	16	210.2	33.2	6.98	6.07	1114	116.0	115.4	2.5	250.0	500	31.25	30.86	30.41	47.02	32.0	8.14	4.50	55.3

FIG. 6.

The output of the pump was obtained from readings of a calibrated oil meter and checked by weights and a second oil meter. Pressure gauges were installed on the suction and discharge lines as near the pump as possible. Thermometers were located in suction and discharge lines adjacent to the pump to indicate the rise in temperature of the oil across the pump. The speed of the pump was determined by a direct reading revolution counter and a stop-watch.

DATA SHEET OF SPEED & PRESSURE-CAPACITY TESTS.

No.	Time	Motor				Oil Pressure		Oil Temperature		Primary Meter		Secondary Meter	
		Watts	Amps	Speed		Suction	Discharge	Suction	Discharge	Reading	Diff.	Reading	Diff.
	hr.-min.	Volts	Amps	R.P.M.		lbs. per sq. in.		°F.	°F.	Gal.	Gal.	Gal.	Gal.
1	10:42	207	28	1133		1	200	110	112	17847	—	6126	—
	10:46	207	28	1135		1	200	113	114	17983	136	6264	137
	10:50	210	28	1136		1	200	112	114	18120	137	6402	137
	10:54	212	28	1136		2	200	113	114	18256	136	6543	139
	10:58	210	28	1135		2	200	110	112	18393	137	6683	140
	11:02	210	28	1134		2	200	110	112	18531	138	6823	140
2	20 Min.	208.5	28.0	1135		1.5	200.0	111.2	113.0	—	684	—	677
15	1:36	212	38	1078		1.5	300	98	100	31570	—	20780	—
	1:40	212	37	1079		1.5	300	98	100	31690	116	20961	181
	1:44	212	38	1079		1.5	300	100	100	31806	116	20982	121
	1:48	212	38	1074		1.5	300	100	100	31922	116	21103	121
2	12 Min.	212.0	38.0	1075		1.5	300.0	99.0	100.0	—	348	—	363
17	2:26	210	52	1024		0	400	106	110	32006	—	21744	—
	2:30	207	52	1027		0	400	106	110	32094	88	21837	93
	2:34	207	52	1036		0	400	107	110	32182	88	21930	93
	2:38	207	52	1035		0	400	106	108	32267	87	22023	93
2	12 Min.	208.5	52.0	1031		0	400.0	106.3	107.6	—	263	—	279

FIG. 7.

Figure 5 is a summary showing results of the different test runs. Figure 7 is a data run selection from tests at different pressures to show the consistent performance of the pump. The latter results have not been corrected for meter errors.

Figures 8 and 9 show the speed—capacity tests at different pressures before and after applying temperature of oil correction.

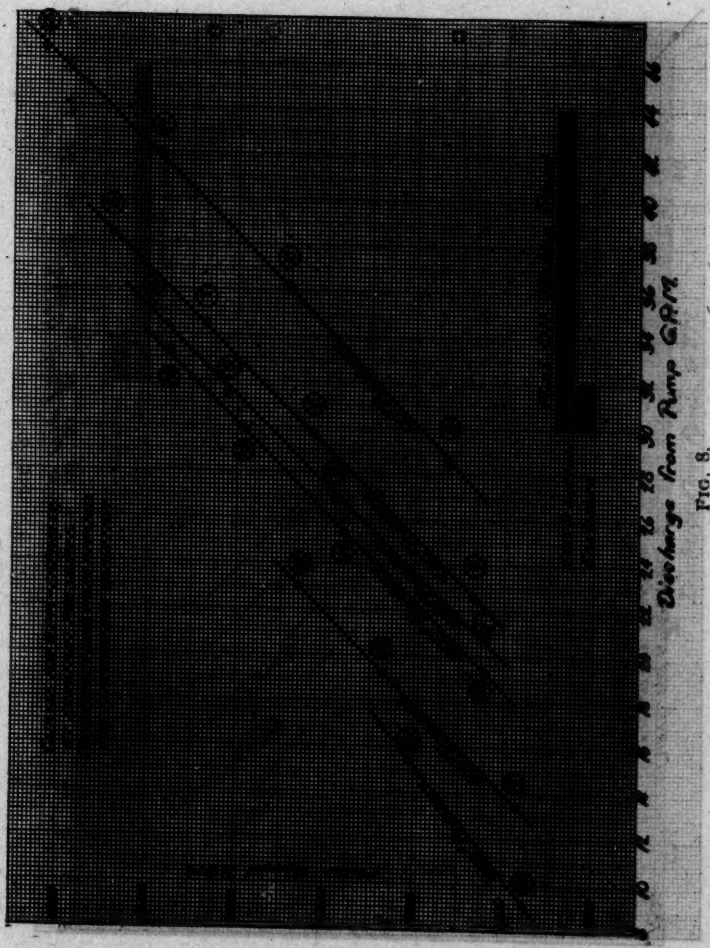


FIG. 8.

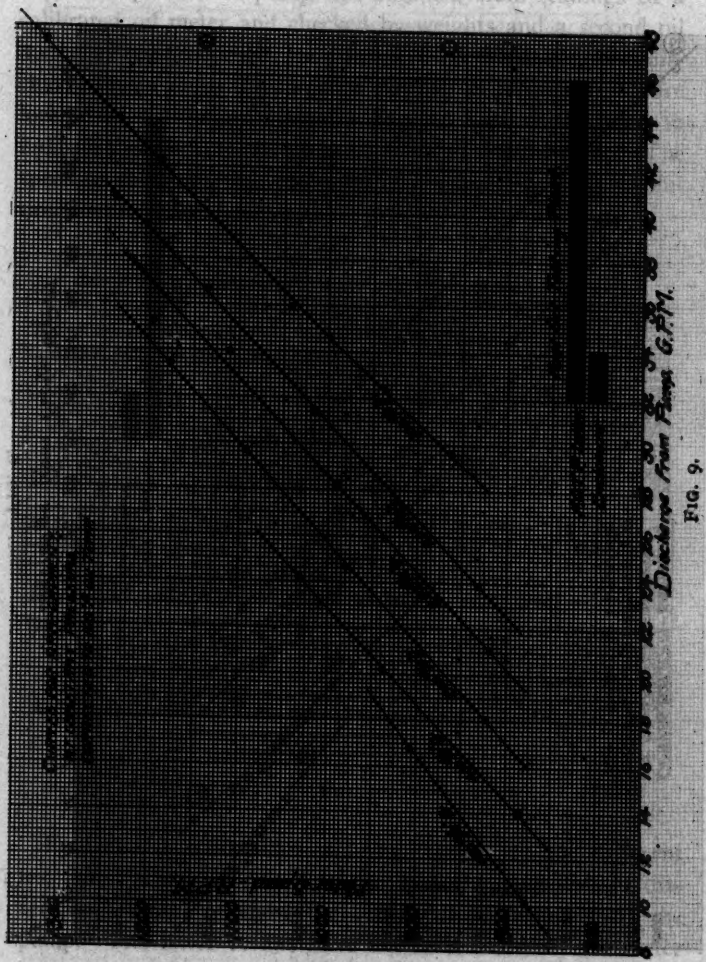


FIG. 9.

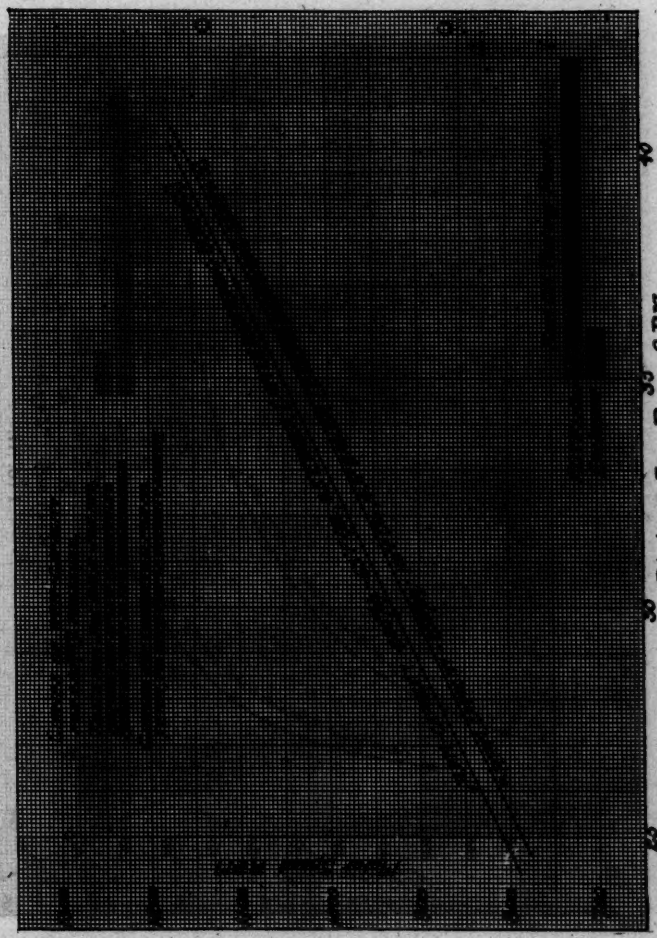


FIG. 10.

FIG. 9.

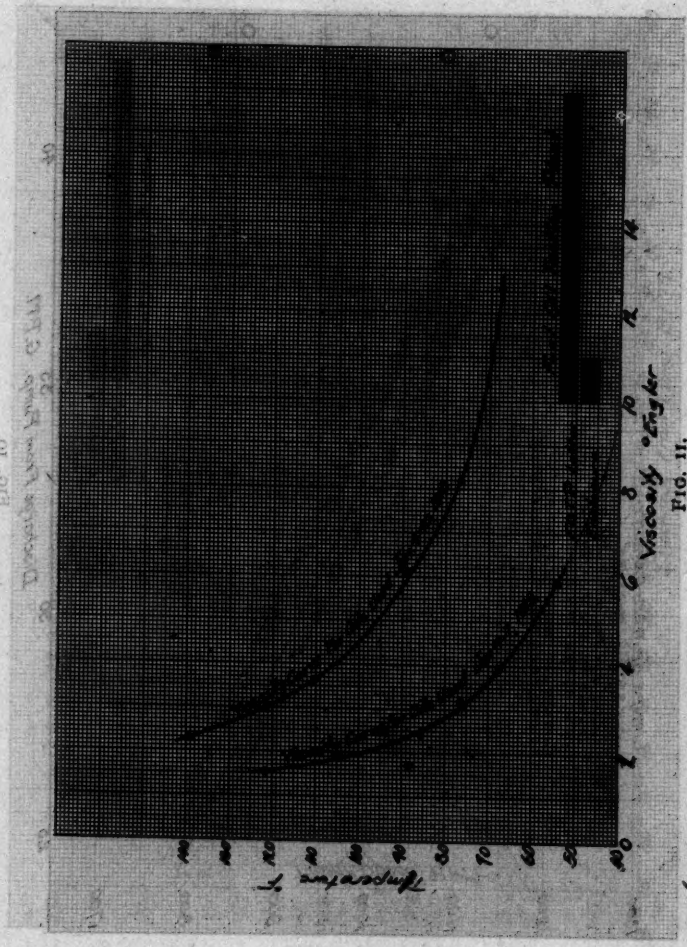


FIG. 11.

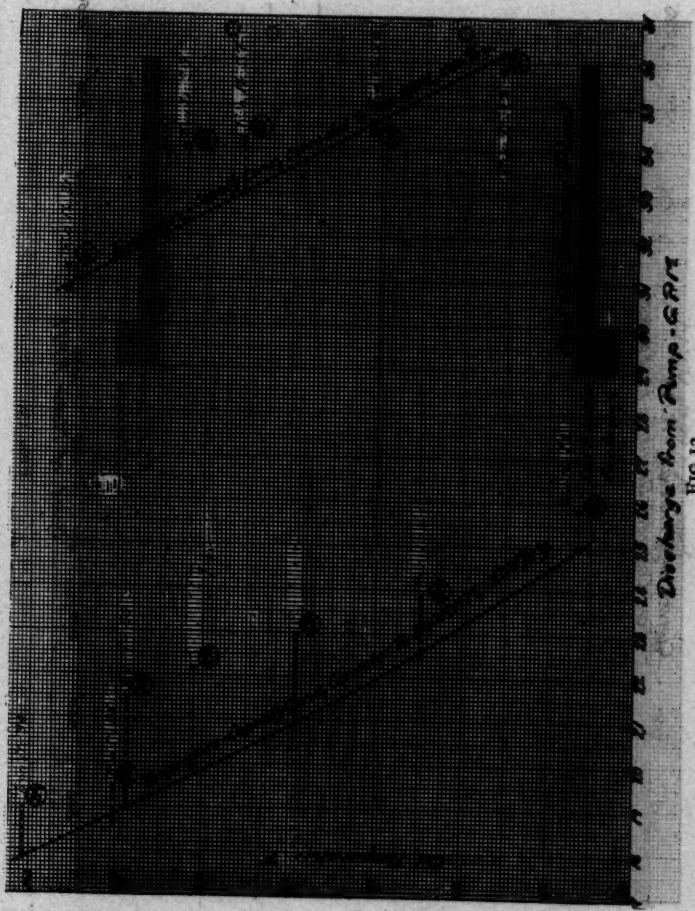


FIG. 12.

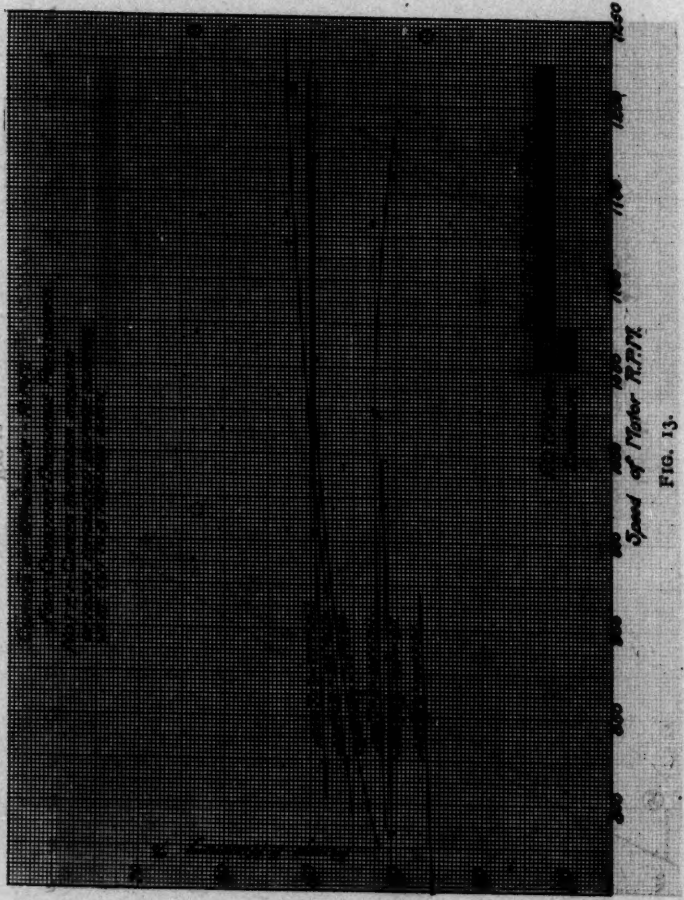


FIG. 13.

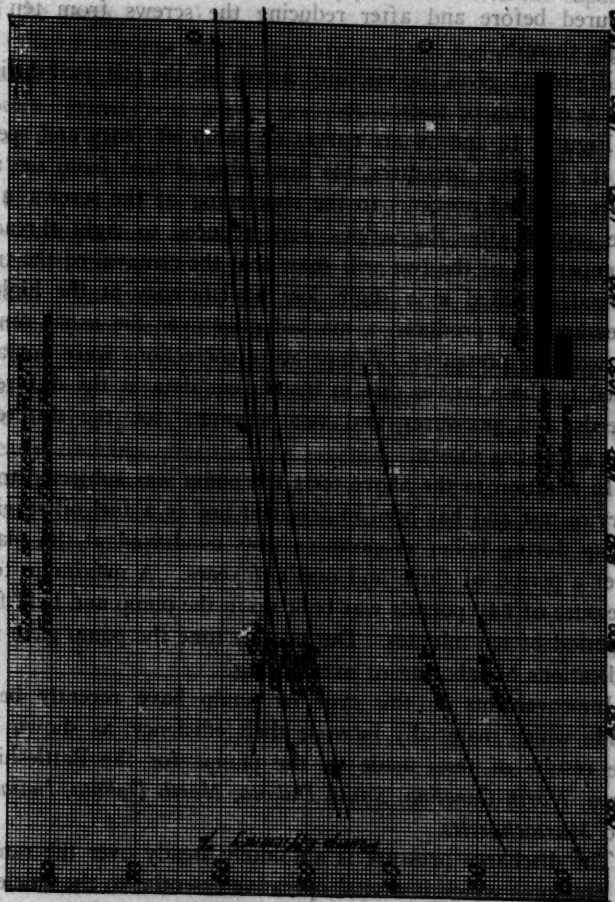


FIG. 14.

FIG. 13.

Figure 10 shows comparative curves for speed—capacity at the same pressure with oil of the same viscosity from results secured before and after reducing the screws from ten to eight threads.

Figure 11 gives the viscosity curves for the oils used during the two series of tests.

Figures 13 and 14 are curves of pump efficiency and speed at the different pressures, the first being plotted from the results obtained previous to the shortening of the screws, and the second after the screws were reduced to eight threads. Comparison of the two will show that the removal of the two threads resulted in a slight loss of efficiency at the higher pressures but produced a marked gain over the usual range of working pressures. Temperature corrections were not applied to these curves due to insufficient data at the different speeds and pressures and this accounts for the different characteristics shown.

In addition to the above tests the pump was subjected to a service test and has been in frequent use as a fuel oil service pump during boiler evaporative tests conducted at the plant over a period of three and one-half years. A maximum test pressure of 540 pounds was built up in the pump and held for a period of ten minutes. During this time the pump gave no indications of laboring or overheating.

Improvements in this type of pump have recently been made which materially increase their efficiency, as shown by the results of tests made on the pumps for the Battleships *Iowa, North Carolina, South Dakota, North Dakota, Indiana* and *Massachusetts.*

The discharge pressure is absolutely steady over the range of pressures which is a vital operative essential to the efficient burning of fuel oil. No air banks or chambers are necessary on the discharge lines. A vacuum of 26 inches is easily obtained upon the closing of the suction valve. The pump has never lost its suction nor failed to pick it up on starting when

the viscosity of the oil being pumped has been such as to allow a velocity of flow in the suction pipe sufficient to the capacity of the pump.

Oils of varying specific gravity ranging from 10.8 degrees Baumé (0.9944 specific gravity) to 32 degrees Baumé (0.8654 specific gravity) have been pumped with no other adjustment than the setting up or slackening off of stuffing box glands.

The pump has required no repairs during the three and a half years of its use other than the occasional renewal of the soft packing in the shaft stuffing boxes and requires no attention during working periods except the oiling of driving gears. The wear on the pump after the above period of service is not perceptible.

The pump is very compactly and rigidly made and is noiseless during operation except for the slight hum occasioned by the driving gears. These characteristics together with the steady discharge pressure and readiness of taking suction particularly adapt this type of pump to the conditions existing on board Naval vessels which require an efficient and reliable fuel oil service pump.

MACHINING PROJECTILE FORGINGS.

NAVY YARD, MARE ISLAND.

By Commander F. J. Cleary, U. S. Navy, Member,

and

Quarterman Machinist F. P. Kruse.

The following description of machining 3-inch and 6-inch projectile forgings, it is believed, will be of interest because of the low manufacturing costs obtained. While the number of projectiles machined was minute compared with war expenditures, the plant produced maximum output for minimum expense. It was a complete, well balanced unit with no lost motion, capable of a tremendous increase of output by the simple process of duplicating the units.

In September, 1915, the Bureau of Ordnance addressed an inquiry to the Navy Yard, Mare Island, requesting estimates of time and cost for machining

10,000 6-inch forged steel common projectiles,
40,000 3-inch forged steel common projectiles,

the rough forgings and rough rotating bands to be furnished by the Bureau.

The Mare Island estimates were as follows:

10,000 6-inch forged steel common projectiles:	
Labor	$42,733.00
Indirect	11,367.00
Material	1,800.00
Total	$55,900.00
40,000 3-inch forged steel common projectiles:	
Labor	$54,588.00
Indirect	15,012.00
Material	1,800.00
Total	$71,400.00

These estimates were based on the actual cost of some small orders for projectiles which had been previously undertaken at Mare Island and contemplated handling the projectiles in the machine shops where the usual line of work was in progress.

On the basis of these estimates the Bureau allotted to Mare Island 10,000 6-inch and 40,000 3-inch projectiles.

These orders were completed and all projectiles accepted and shipped. The total actual costs were:

10,497 6-inch forged steel common projectiles:

Labor	$22,045.72
Indirect	6,127.57
Material	4,615.86
Total	$32,789.15

Unit Navy Yard cost per projectile ..... $ 3.12
Unit cost rough projectile, base plug and band $11.36

41,467 3-inch forged steel common projectiles:

Labor	$10,563.60
Indirect	2,940.83
Material	1,231.57
Total	$14,736.00

Unit Navy Yard cost per projectile $0.36
Unit cost rough projectile and band $2.18

These labor, indirect and material costs include the manufacture of all necessary tools, jigs and gauges, grommets and canvas protecting bands, painting the projectiles, the manufacture of a stamping press for stamping the bases of the projectiles, the manufacture of a double splindle boring machine for boring the 6-inch projectiles and complete overhaul of all machine tools (after the completion of the work), putting them in first-class condition for any other work.

The difference between the actual and estimated costs is due to the fact that a careful study was made of the subject, and that the layout of machines, the design of tools, jigs and

fixtures; the routing of the work and the handling of material and personnel were the results of intelligent thought, thorough preparation and coordination.

Before the receipt of any material it was decided to assemble all necessary tools in a separate building thus making the projectile shop an independent activity.

Three special machine tools were provided. These were a hydraulic press for banding 3-inch and 6-inch projectiles, purchased from the Rochester West Tire Setter Company; a stamping press designed and built at the Navy Yard, and a double spindle turret lathe for boring the cavities of the 6-inch projectiles, also designed and built at the Navy Yard. This special purpose turret lathe was a necessity for the reason that there were only two machines in the machine shops capable of the work of boring the 6-inch forgings; one, an old single spindle Bement-Pond turret lathe; the other, a new 84-inch Cincinnati Bickford radial drill. The Pond turret lathe was needed for finish facing, reaming and counterboring the base and turning and forming the copper rotating bands of the 6-inch projectiles. The 84-inch radial was the only radial in the shops capable of driving a high speed drill and as its use averaged not less than six hours per day (8-hour day) it could not be considered available. With these exceptions all machine tools used were selected from the machine shops and fitted with turret tool heads, radius turning attachments, knurling devices, expanding chucks, pot chucks, limit stops, etc. An oil fuel furnace was built for heating rotating bands.

The cutting tools, all of high speed steel, were standardized for each operation of each machine tool—the operators not being permitted to grind or to change the shape of any tool.

Spare sets of boring bars, reamers and all other cutting tools were kept at each machine. Used cutting tools were collected daily and reground tools distributed, making it unnecessary for the operators to leave their machines.

In addition, where parts of machine tools were known to be weak, spares were kept on hand so that the lost time due to breakdowns was reduced to a minimum. With one or two exceptions, the men employed were selected from helpers employed in the various trades in the Navy Yard. With proper teaching and by the use of limit stops and gauges, men with very little previous knowledge of machinery developed into good operators in a week and were capable of producing the maximum output from their machines at the end of one month.

Owing to the "leave with pay" situation, it was necessary to carry several additional men on the shop rolls. These men, while usually engaged in other work, were all trained as substitutes at some of the machine tools, so that a relief operator was always available if the regular operator was absent for any reason. In this way each machine tool produced a full day's output every day.

Continuous tables were built at the height of the work level of the machines, obviating the lifting of work from the floor at each machine; swinging davits with tongs and lever and other labor saving devices were provided; machine tools were placed in order of sequence of operation; the rough projectiles entered one end of the shop and the finished product was shipped from the other end. By these methods all the operators' energy and time were devoted to productive labor only and no time was lost or confusion caused by supply lines to machine tools crossing each other.

The 3-inch and 6-inch projectiles were machined at the same time, the 6-inch projectiles being handled on the east side of the shop, the 3-inch on the west side. The average number of men engaged in the manufacture of 6-inch projectiles, including base plugs, banding, stamping, assembling, applying rope grommet and canvas protecting band and shipping was 14; of these, 11 were employed in machine operations. The average output per eight-hour day was 60 finished projectiles.

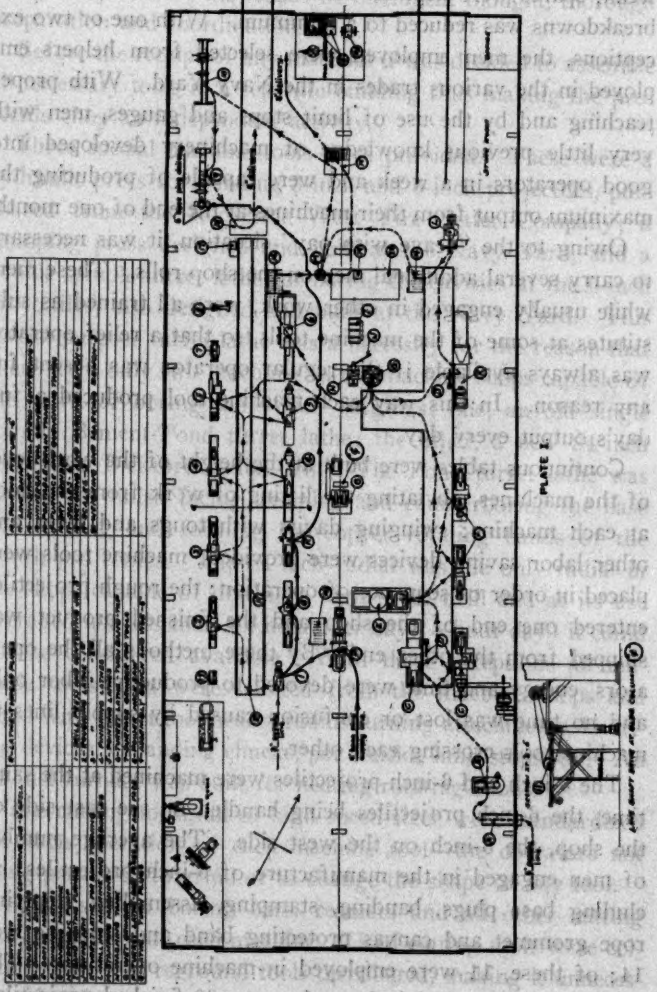


PLATE I.

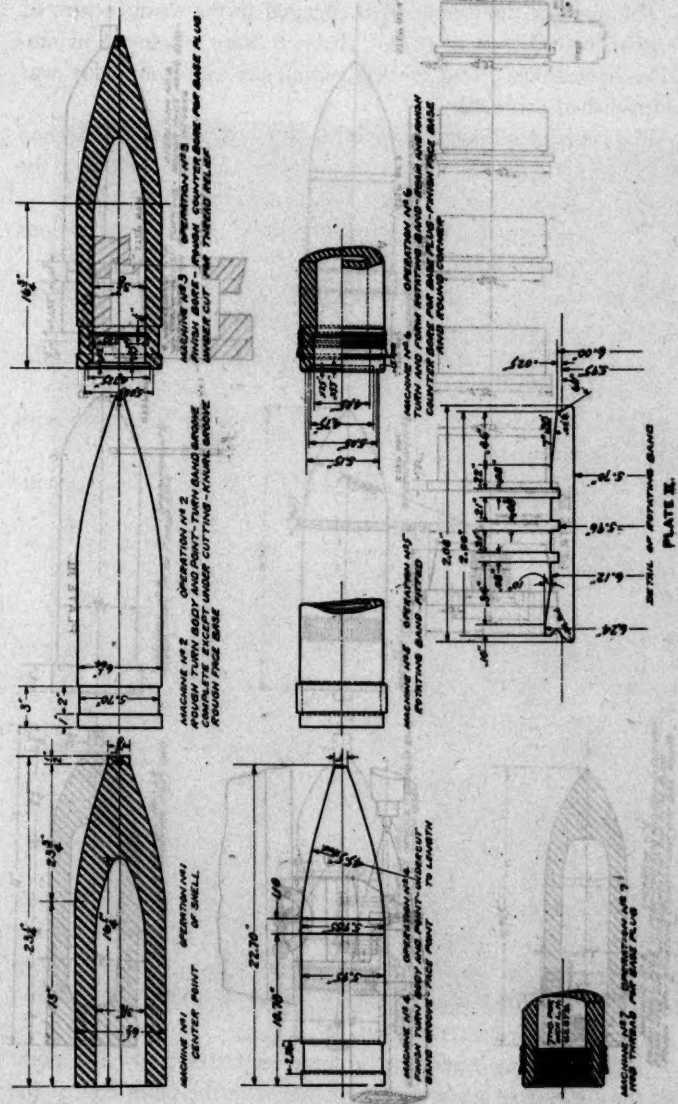


PLATE II.

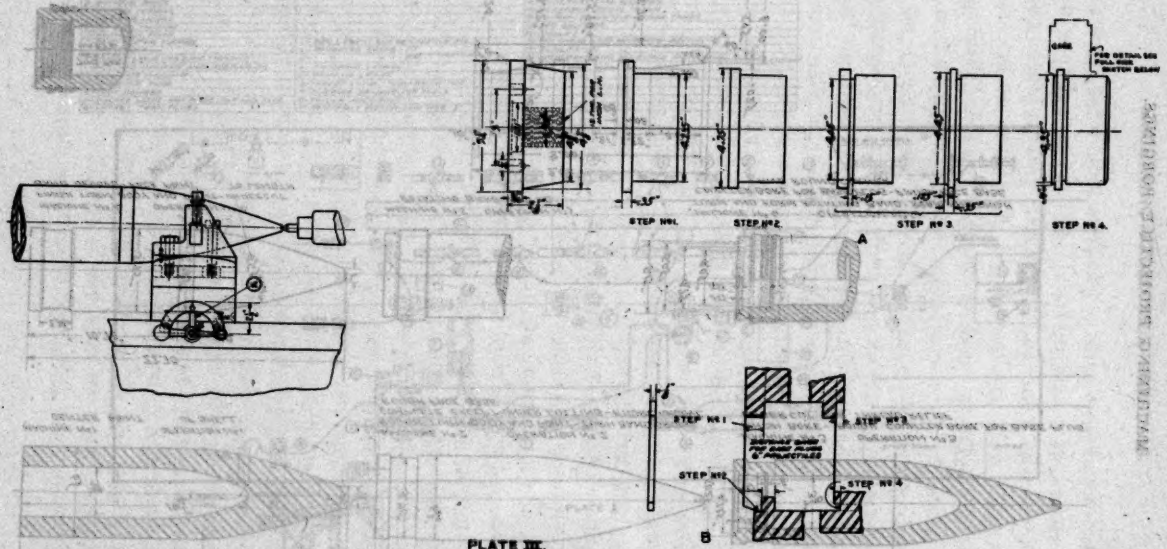

STEP No1.
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PLATE III.

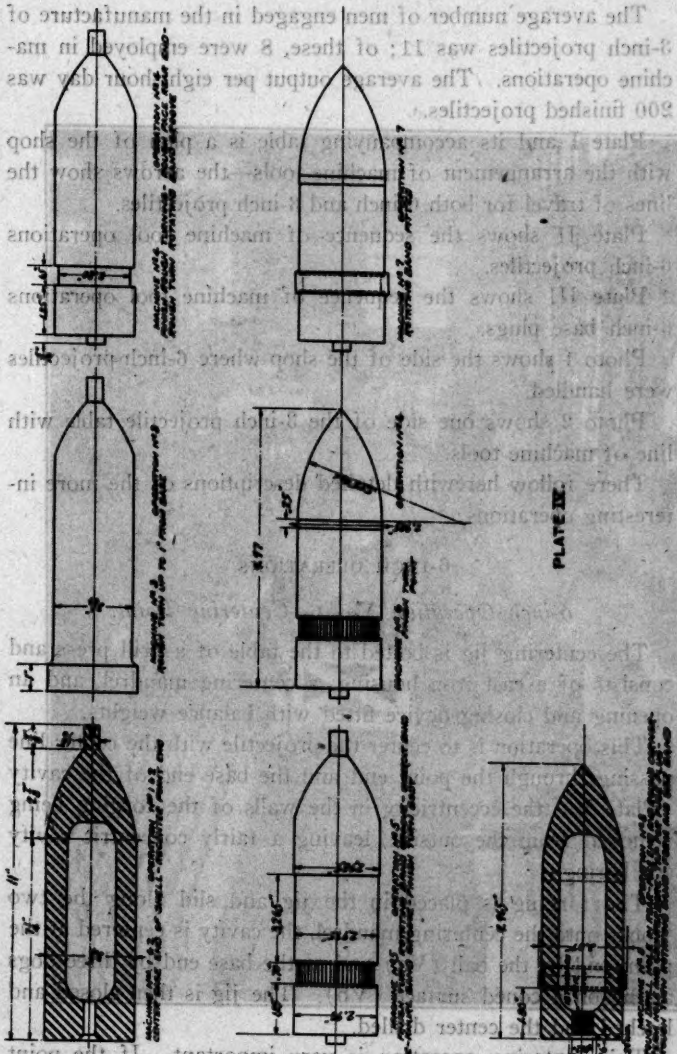


PLATE III.

The average number of men engaged in the manufacture of 3-inch projectiles was 11; of these, 8 were employed in machine operations. The average output per eight-hour day was 200 finished projectiles.

Plate I and its accompanying table is a plan of the shop with the arrangement of machine tools—the arrows show the lines of travel for both 6-inch and 3-inch projectiles.

Plate II shows the sequence of machine tool operations 6-inch projectiles.

Plate III shows the sequence of machine tool operations 6-inch base plugs.

Photo 1 shows the side of the shop where 6-inch projectiles were handled.

Photo 2 shows one side of the 3-inch projectile table with line of machine tools.

There follow herewith detailed descriptions of the more interesting operations:

6-INCH OPERATIONS.

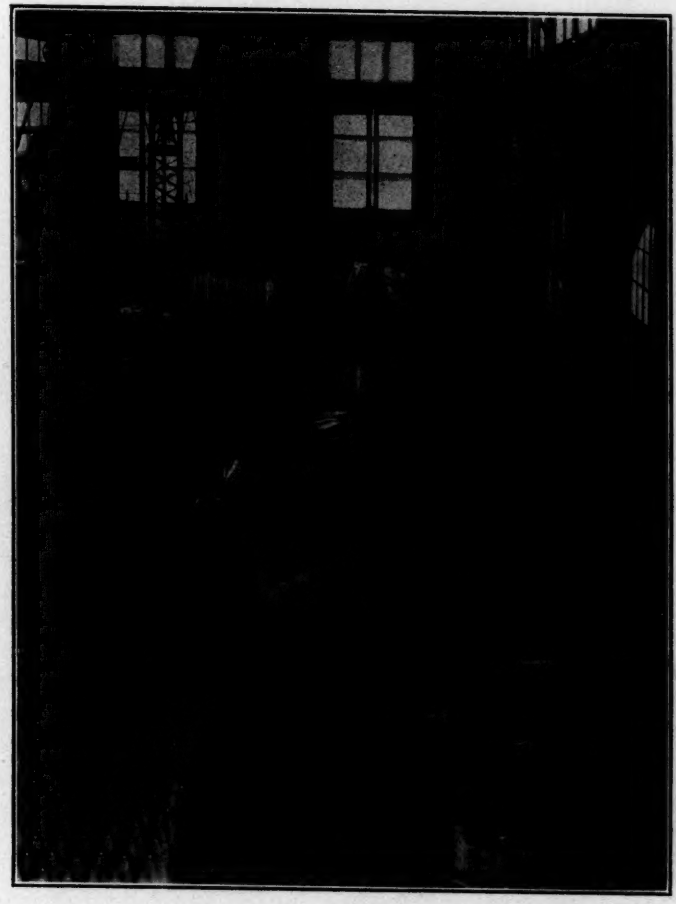
6-inch Operation No. 1—Centering Point.

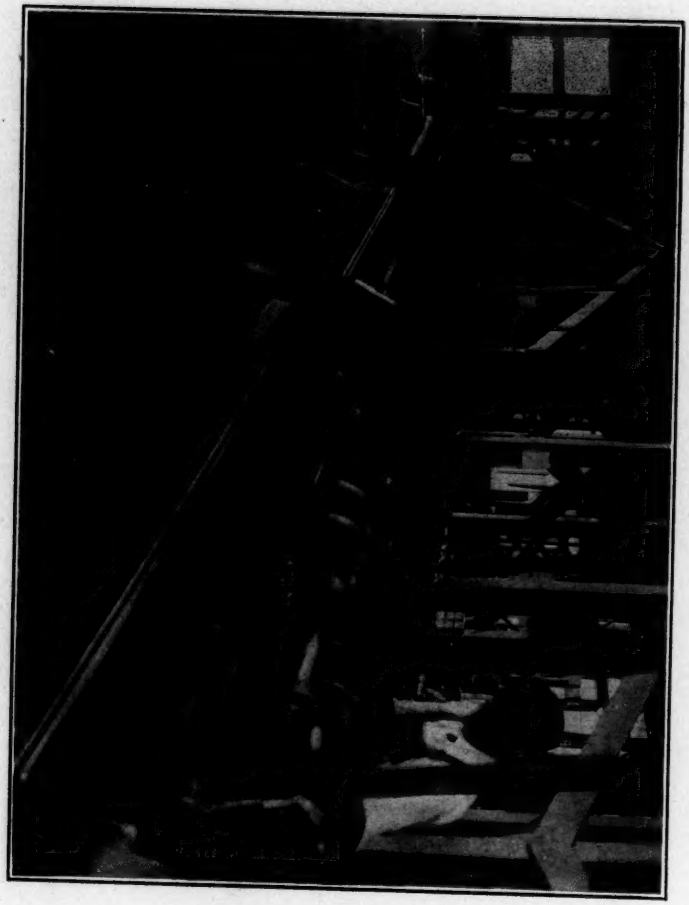
The centering jig is bolted to the table of a drill press and consists of a cast iron housing, a centering mandrel, and an opening and closing device fitted with balance weights.

This operation is to center the projectile with the center line passing through the point end and the base end of the cavity (Plate V), the eccentricity in the walls of the forging being removed from the outside, leaving a fairly concentric cavity for boring.

The forging is placed in the jig and slid along the two tracks onto the centering mandrel, the cavity is centered at the point end by the ball (Va) and at the base end by three dogs acting on a coned surface (Vb). The jig is then closed and latched and the center drilled.

This centering operation is very important. If the point of the forging is centered out of line with the point end of the





cavity it makes the boring difficult and will cause eccentricity in the walls of the finished projectile. The centering jig was checked for perfect alignment not less than twice each week. This operation was performed by one man—cost per forging $.0068.

6-inch Operation No. 2—Rough Turning.

For rough turning, two 48-inch Niles lathes were fitted with expanding mandrels, turret tool heads, knurling attachments, radius forming tracks, and special tool heads for radius turning tools. In this operation the forging is rough turned to 6 1/64 inches diameter, the base rough faced and the rotating band groove machined to size and knurled. The band groove is not undercut in this operation.

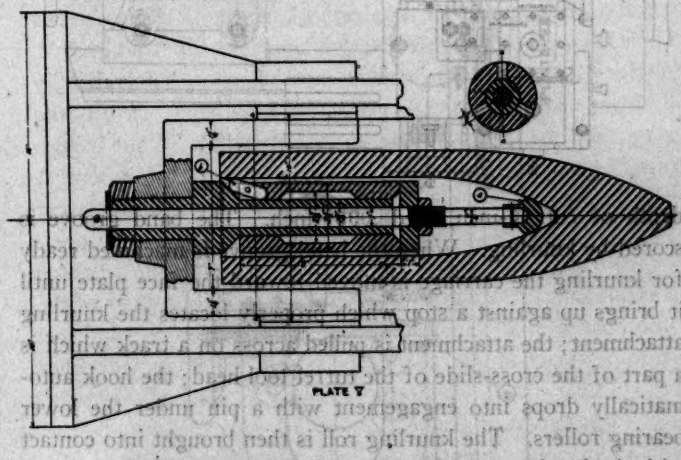
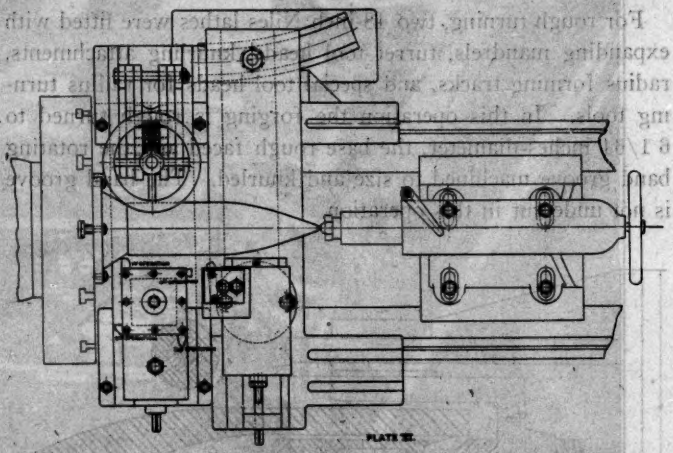


PLATE V

A forging is placed on the expanding mandrel by the use of the davit, swinging lever and tongs. Thirty seconds is the average time used in removing a machined forging and chucking a rough forging.

The turret tool head (Plate VI) is provided with tools for four sub-operations: First, turn body (radius tool turns

ogival); second, rough face base; third, cut band groove to depth; fourth, square sides of band groove to minimum width. The forgings are rough turned to the required diameter in two cuts; the cylinder turning and ogival turning tools are set at a fixed distance so that both tools commence and finish cutting at the same time. The cutting speed is 50 feet per minute,



depth of cut ⅛-inch, feed .0937-inch. The band groove is scored by knurling. When the band groove is machined ready for knurling the carriage is moved toward the face plate until it brings up against a stop which properly locates the knurling attachment; the attachment is pulled across on a track which is a part of the cross-slide of the turret tool head; the hook automatically drops into engagement with a pin under the lower bearing rollers. The knurling roll is then brought into contact with the band groove by means of the screw and hand-wheel.

Plate VII shows details of the knurling attachment—the arms to which the knurling roll is fitted (VIIb) and the bottom arms to which the bearing rollers and hooking pin are attached are hinged (VIIc), so that there is no pressure on the lathe centers at any time.

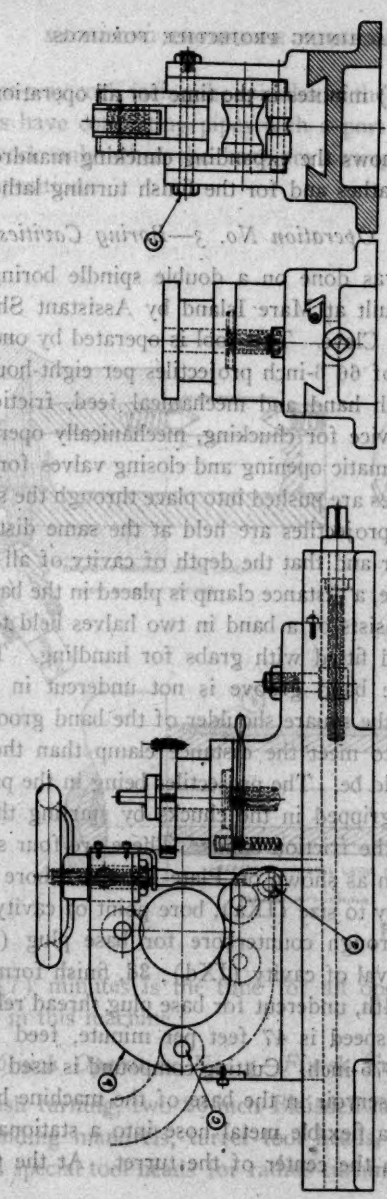


PLATE XII

Nineteen (19) minutes is the time for all operations in these lathes.

Plate VIII shows the expanding chucking mandrels for the rough turning lathes and for the finish turning lathes.

6-inch Operation No. 3—Boring Cavities.

This work was done on a double spindle boring machine designed and built at Mare Island by Assistant Shop Superintendent C. T. Cleve. This tool is operated by one man and has a capacity of 66 6-inch projectiles per eight-hour day. It is equipped with hand and mechanical feed, friction locking and opening device for chucking, mechanically operated quick return and automatic opening and closing valves for lubricant.

The projectiles are pushed into place through the spindle. To insure that all projectiles are held at the same distance from the turret center and that the depth of cavity of all projectiles shall be the same, a distance clamp is placed in the band groove. This device consists of a band in two halves held together by coil springs and fitted with grabs for handling. This is the reason that the band groove is not undercut in the rough turning lathes, the square shoulder of the band groove being a better surface to meet the distance clamp than the undercut knife edge would be. The projectiles being in the proper position, they are gripped in the chucks by starting the machine and tightening the friction bands. There are four steps in the boring operation as shown on Plate IX: 1st, bore cylindrical portion of cavity to size (IXa), bore point of cavity to proper depth (IXb), rough counterbore for base plug (IXc); 2d, rough form ogival of cavity (IXd); 3d, finish form ogival of cavity (IXe); 4th, undercut for base plug thread relief (IXf).

The cutting speed is 47 feet per minute, feed .0625-inch, depth of cut .1875-inch. Cutting compound is used and is supplied from a reservoir in the base of the machine by a rotary pump through a flexible metal hose into a stationary vertical receiver pipe in the center of the turret. At the top of this

pipe is a one-way cock facing the spindles of the machine. Each set of bars have connecting pipes with a port which connects with the cock only when the bars are swung around to the operating position.

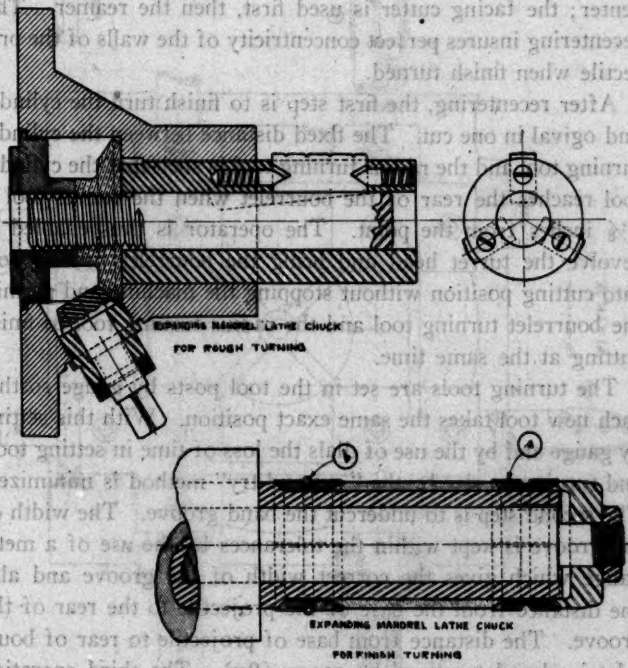


PLATE VIII.

Seven (7) minutes is the time for all operations on two projectiles in this machine.

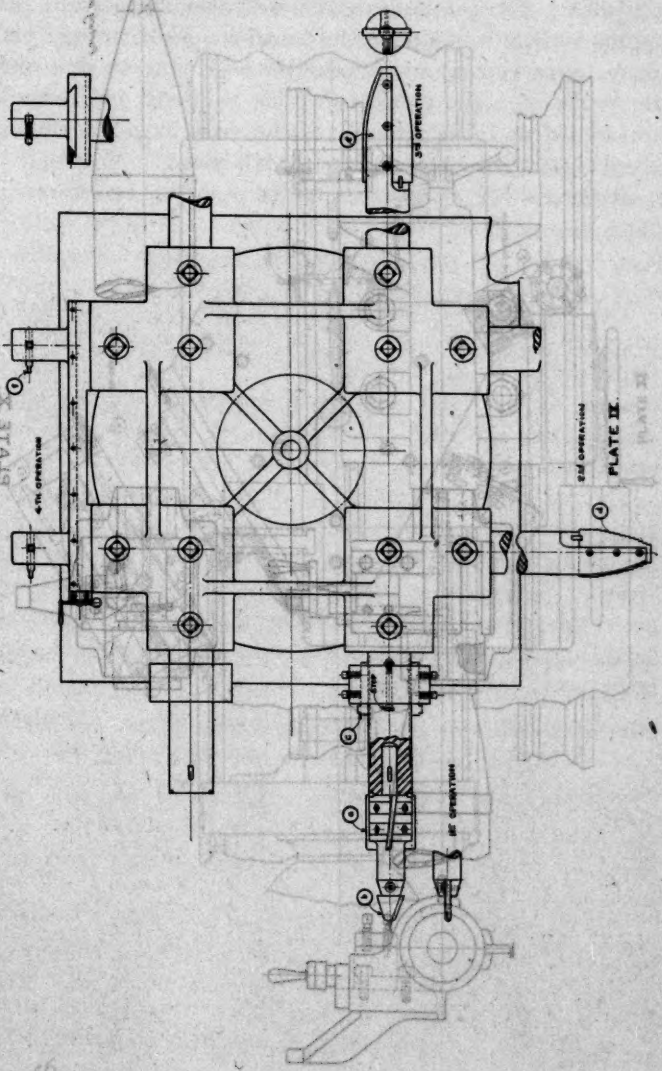
6-inch Operation No. 4—Finish Turning.

For finish turning, two 20-inch Blaisdell lathes were fitted with expanding mandrels, turret tool heads, radius forming tracks and special tool heads for radius turning tools.

Due to the heavy strain on the drilled center of the projectile during the rough turning, it was found necessary to face and re-ream the original center. This is done after the projectile is chucked in the finishing lathes, by means of a tool holder made to fit the tail stock spindle without removing the center; the facing cutter is used first, then the reamer. This recentering insures perfect concentricity of the walls of the projectile when finish turned.

After recentering, the first step is to finish turn the cylinder and ogival in one cut. The fixed distance between the cylinder turning tool and the radius turning tool is such that the cylinder tool reaches the rear of the bourrelet when the radius tool is 1⅛ inches from the point. The operator is thus enabled to revolve the turret head and bring the bourrelet turning tool into cutting position without stopping the machine and permits the bourrelet turning tool and the radius turning tool to finish cutting at the same time.

The turning tools are set in the tool posts by gauge so that each new tool takes the same exact position. With this setting by gauge and by the use of dials the loss of time in setting tools and turning to size by the "cut-and-try" method is minimized. The second step is to undercut the band groove. The width of the groove is kept within the tolerances by the use of a metal gauge which gives the correct width of the groove and also the distance from the base of the projectile to the rear of the groove. The distance from base of projectile to rear of bourrelet is also determined by gauge (9g). The third operation is to cut the projectile to length and form the point. A combined ogival rest and tool holder is used (Plate X). This fixture is a cast iron body with a foot rest, is machined to slip on the tail-stock spindle, is fitted with a revolving bushing of bearing metal formed to fit the ogival of the projectile, and has a tool post operated by a hand screw. When the second step is completed the tail stock is released and pushed back, the fixture adjusted to the tail-stock spindle, and the tail stock


3d OPERATION
4TH OPERATION
2nd OPERATION
1st OPERATION
PLATE II.

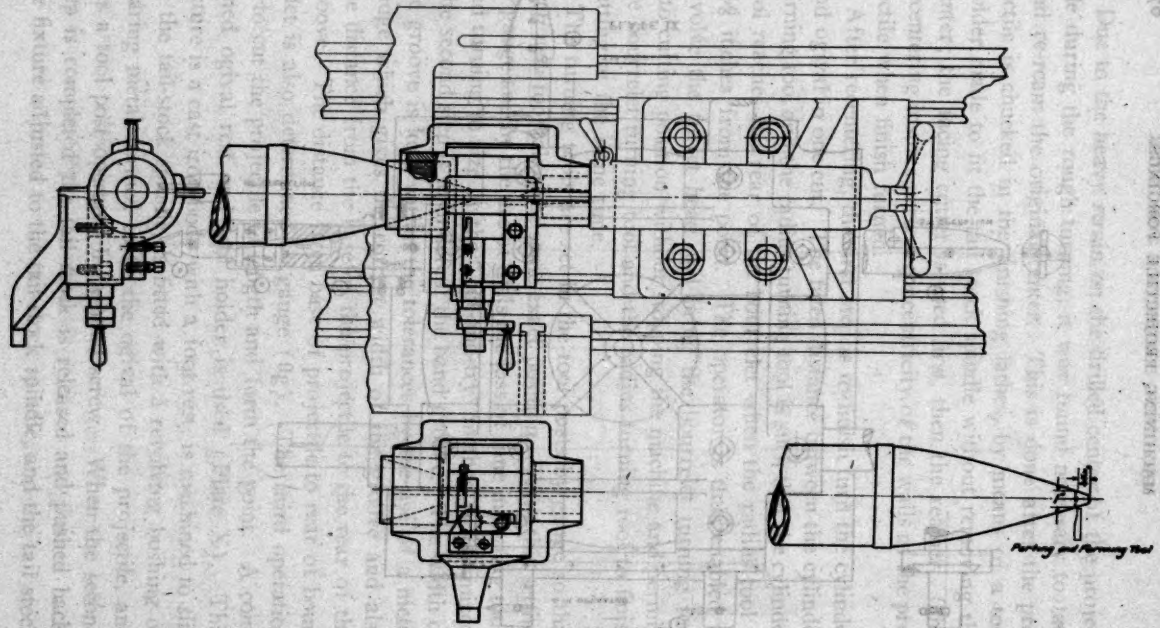


PLATE X.

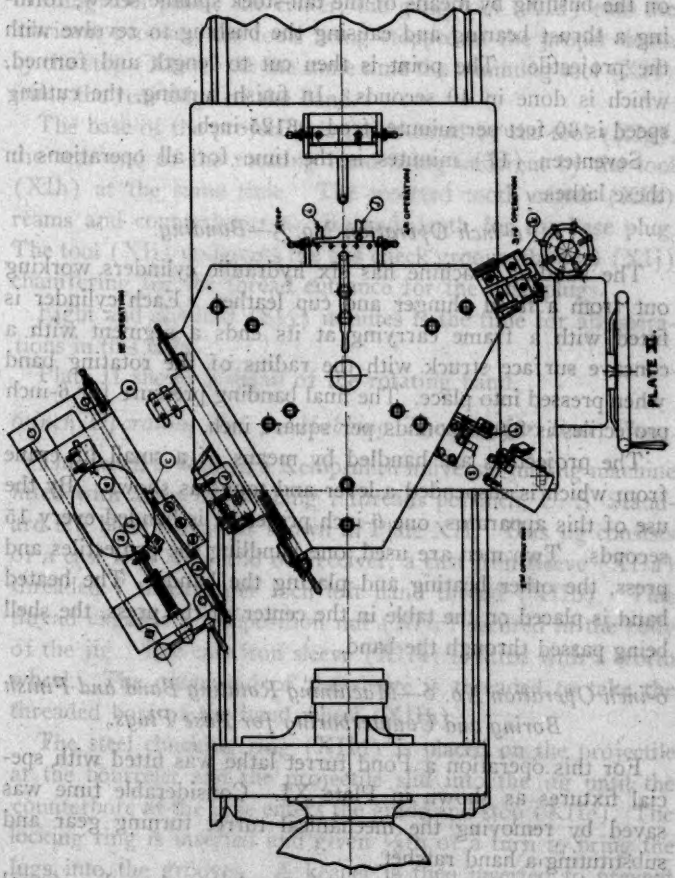


PLATE XI.

brought forward until the bushing comes in contact with the projectile. The tail stock is then secured and a pressure put on the bushing by means of the tail-stock spindle screw, forming a thrust bearing and causing the bushing to revolve with the projectile. The point is then cut to length and formed, which is done in 40 seconds. In finish turning, the cutting speed is 60 feet per minute, feed .03125-inch.

Seventeen (17) minutes is the time for all operations in these lathes.

6-inch Operation No. 5—Banding.

The banding machine has six hydraulic cylinders working out from a fixed plunger and cup leather. Each cylinder is fitted with a frame carrying at its ends a segment with a concave surface struck with the radius of the rotating band when pressed into place. The final banding pressure for 6-inch projectiles is 2,000 pounds per square inch.

The projectiles are handled by means of a small jib crane from which is suspended a lever and tongs as shown. By the use of this apparatus, one 6-inch projectile is banded every 15 seconds. Two men are used, one handling the projectiles and press, the other heating and placing the bands. The heated band is placed on the table in the center of the press, the shell being passed through the band.

6-inch Operation No. 6—Machining Rotating Band and Finish Boring and Counterboring for Base Plugs.

For this operation a Pond turret lathe was fitted with special fixtures as shown in Plate XI. Considerable time was saved by removing the mechanical turret turning gear and substituting a hand ratchet.

The machining of the rotating band was done by several tools fitted into a frame sliding on the turret head and moved at right angles to the axis of the projectile by a screw. The band is rough turned by the two tools (XIa), the depth of cut being governed by the frame coming up against the stop

(XIb). The movement of the frame is then reversed and the band is finish turned by the forming tool (XIc). After the forming tool passes the center, the gang tool (XId) cuts the lubricant grooves, the tools being stopped at the proper depth by the stop (XIe). At the same time the trimming tool (XIf) trims the rear edge of the band.

The base of the projectile is finish faced by the tool (XIg), the rear end of the rotating band being undercut by the tool (XIh) at the same time. The inserted tooth cutter (XIk) reams and counterbores to size and depth for the base plug. The tool (XIi) undercuts the gas check groove, the tool (XIj) chamfering for the thread entrance for the base plugs.

Eight and one-half (8½) minutes is the time for all operations in this lathe.

Plate II shows a detail of the rotating band.

6-inch Operation No. 7—Hobbing Threads for Base Plugs.

This work is done in a Kempsmith universal milling machine fitted with a jig for hobbing 7 threads per inch, U. S. Standard left hand thread as shown in Plate XII. This jig consists of a cast iron body and oil receiver, a cast iron sleeve (XIIa) threaded 7 threads per inch left hand thread (XIIb). This thread engages a composition nut (XIIc) secured in the body of the jig. The cast iron sleeve (XIIa) is fitted with a worm wheel. The outer end of the sleeve is threaded to take the threaded boss of the hand wheel (XIIh).

The steel chucking ring (XIId) is placed on the projectile at the bourrelet and the projectile slid into the jig until the counterbore at the base enters the guide and stop (XIIe). The locking ring is inserted and given ⅛th of a turn to bring the lugs into the grooves. A keeper is then inserted to prevent the locking ring turning independently of the hand wheel. The hand wheel is turned to the left, locking the projectile into the sleeve and accurately setting it into the jig by the base end and the bourrelet. The driving worm is mounted on an

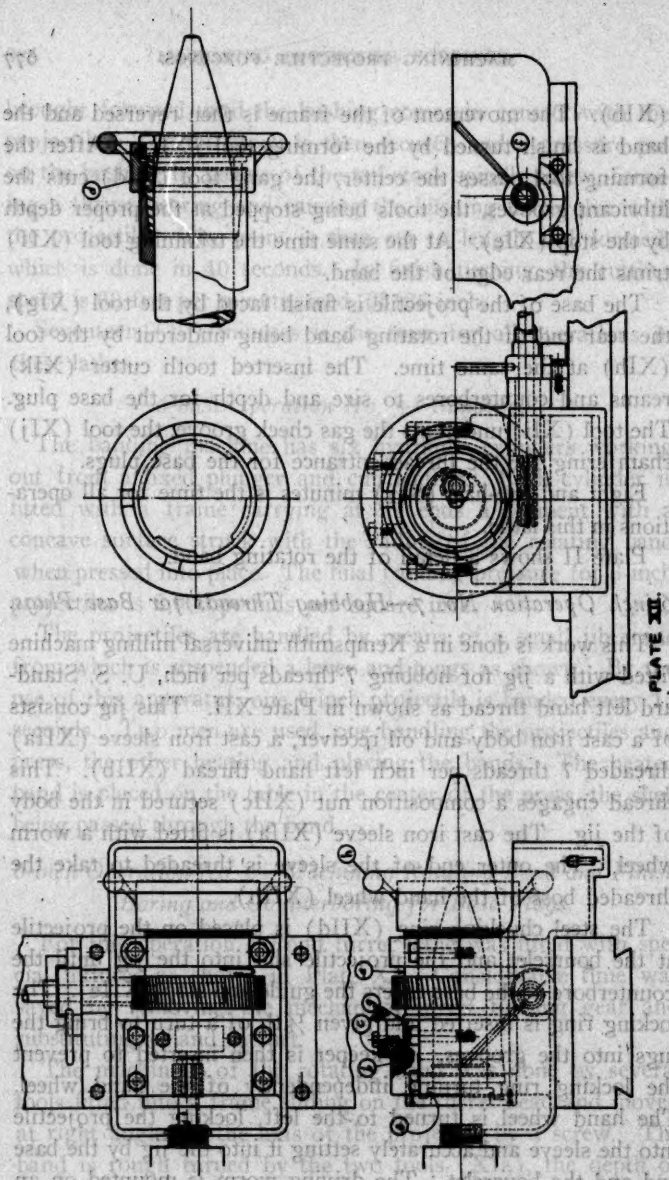


PLATE XII.

eccentric sleeve (XIIf) actuated by a lever. The worm wheel and worm are disengaged, the hand wheel again turned to the left until the worm wheel brings up against the face of the housing (XIIg). The worm wheel and worm are again engaged and the table and jig moved up against a stop until the hob has been brought to the required depth of thread. The hob is a series of annular cutting teeth, 60 degrees—7 threads to the inch, U. S. Standard left hand. The hob is of sufficient length to cover the full length of the thread in the base of the projectile; as the projectile moves from the hob 1/7 of an inch per revolution, one revolution of the projectile cuts a thread the entire distance.

As it was desirable to make all base plugs interchangeable, it was found necessary to make two revolutions of the projectile; by so doing the threads were perfect in size and form. After the projectile is removed from the fixture, an adjustable taper tap accurately set to size is run through the threads to remove burrs, etc., and as a gauge for size.

Eight and one-half (8½) minutes is the time for hobbing the thread (two cuts) in one projectile.

6-inch Operation No. 8—Stamping Base.

For stamping the bases of projectiles a small air hammer was built at the Navy Yard. The hammer is raised by air pressure and dropped by gravity. The lettering die is secured in the sliding plate and held in place by a set collar, sufficient clearance being allowed between the collar and the sliding plate to prevent the weight of the hammer from injuring the plate. As it is necessary to lift the sliding plate to remove the projectile, the plate is counterbalanced, being held in any desired position by the two clamps. The anvil block is interchangeable, a different size block being used for each caliber of projectile. The air throttle valve is so constructed that after a little experience, the operator can prevent the rebound of the hammer which would blur the letters.

With this hammer two men stamp 6-inch projectiles at the rate of 120 per hour.

6-inch Operation No. 9—Base Plug Operation A.

An old Potter and Johnson turret lathe was fitted for this work which consisted of drilling, facing, reaming, counterboring, rough turning collar, cutting gas check groove and tapping.

A tungsten steel twist drill is used, the facing tool being used while drilling. The drilled hole is then brought to size and counterbored by the reamer the depth of the counterbore being fixed by the stop. While the reamer is counterboring, the collar is rough turned by the tool. The gas check groove is then undercut by the tool. For tapping the fuse a bastard gun tap is used, 1½ inches diameter, 12 threads, U. S. Standard left hand thread. This tool taps a hole 1¾ inches long, the tolerance of diameter being .004 of an inch. After considerable experimenting this style of tap was found to be very durable and exact, due to the longer surface, the cutting surface being ground and all the teeth being backed off the entire length of the lands.

Six and two-thirds (6 2/3) minutes is the time for all of these operations.

About 200 plugs are run through Operation A; the entire carriage is then reversed bringing into play the facing tool; the plugs are then rechucked but reversed and the front end of the plug faced, bringing the plug to the proper thickness.

6-inch Operation No. 10—Base Plug Operation B.

Base Plug Operation B consists in drilling the two holes for the teat wrench.

A triple Sellew drilling attachment is fitted to the spindle of a drill press. Only two drills are used; in place of the center drill, a telescoping sleeve and spindle are fitted, the bottom of the sleeve having a flange the same size as the counterbore of the base plug. This flange is used for center-

ing the plug. A stop secured to the spindle travels in a slot in the sleeve limiting the depth of the wrench holes. A spiral spring under compression is fitted in the sleeve under the spindle. This spring forces the sleeve flange down into the counterbore before the drills begin to cut; it also thrusts the plug clear of the drills when the spindle of the drill press is raised. This double drilling device eliminates the use of a jig or holding-down clamps.

Three (3) minutes is the time for this operation.

6-inch Operation No. 11—Base Plug Operation C.

Base Plug Operation C consists of 4 steps—finish turning body, finish turning collar, turning thread relief, and turning and dove-tailing gas check groove. This work is performed on a 20-inch American lathe fitted with stops and a turret tool head, as shown in Plate XIII. The plug is centered on a

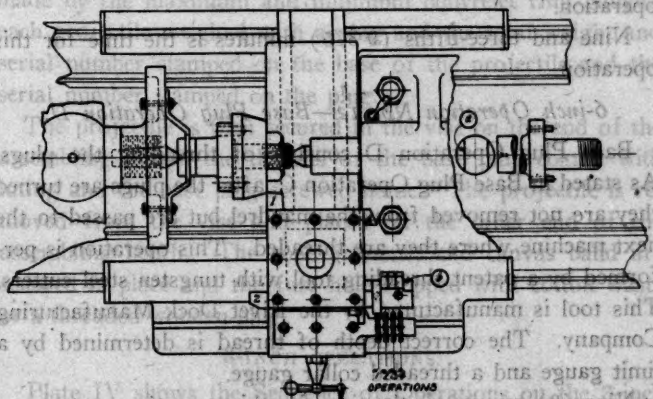


PLATE XIII.

mandrel by its counterbore and driven by two pins in the mandrel which enter the teat wrench holes in the plug. The mandrel has a flat on two sides (XIIIa) by which it is driven as shown.

All diameters are determined by the use of stops (XIIIb) which are fitted and hinged on a plate which is secured to the lathe carriage. On the cross feed is secured another plate with four adjusting screws, 1, 2, 3 and 4, locked in place by headless set screws.

Plate XIII shows tool 1 in position for turning the body with stop 1 in position, stops 2, 3 and 4 being in their upright neutral position. The different steps are shown in Plate IIIA, the correct distances being gauged by a sheet metal gauge (Plate IIIB). By the use of this system of stops and gauges all the steps are performed in succession and the lathe is not stopped until all steps are completed. While the lathe is in operation, the operator fits the next plug to be turned to a mandrel. After these steps are completed the plug and mandrel are removed from the lathe; the plug is not removed from the mandrel, however, until it has been threaded, which is the next operation.

Nine and three-fifths (9 3/5) minutes is the time for this operation.

6-inch Operation No. 12—Base Plug Operation D.

Base Plug Operation D consists of threading the plugs. As stated in Base Plug Operation C, after the plugs are turned they are not removed from the mandrel but are passed to the next machine where they are threaded. This operation is performed by a patent threading tool with tungsten steel cutters. This tool is manufactured by the Rivet Dock Manufacturing Company. The correct depth of thread is determined by a limit gauge and a threaded collar gauge.

Nine (9) minutes is the time for this operation.

6-inch Operation No. 13—Cleaning and Painting.

After the projectiles are stamped they are washed thoroughly clean inside and out in a bath of turpentine substitute and stood on their bases in drain pans to dry, after which the cavities are painted with an approved anti-acid preparation.

For painting the outside a special fixture was used consisting of two T-irons bolted together with distance pieces so that the projectile rested with the rotating band on one T-iron, the bourrelet on the other T-iron. At the center of each T-iron was located two rollers, the tops of which projected slightly above the upper surface of the T-irons.

The unpainted projectiles are placed on one end of the fixture and are rolled along the fixture in succession onto the rollers where the painter revolves the projectile with one hand using the brush with the other.

Two (2) minutes is the average time for painting the cavity and exterior of the 6-inch projectiles.

6-inch Operation No. 14—Fit Base Plug and Secure Rope Grommet and Canvas Protecting Band Over Rotating Band.

After the paint is dry, a final inspection of the projectile is made by the maximum and minimum bourrelet ring gauges, each projectile weighed with a plug and the total weight and serial number stamped on the base of the projectile and the serial number stamped on the plug.

The projectile is then secured in the vise on the end of the assembling bench, the threads of the base plug coated with luting and the base plug screwed home. The projectile is removed from the vise and placed on the other end of the assembling bench. The rope grommet and canvas band are secured in place and the fuse hole stopped with cotton waste or a threaded wooden plug.

3-INCH OPERATIONS.

Plate IV shows the Sequence of Operations on the 3-inch projectiles. A better understanding of the method used in centering the 3-inch forgings will be had if some of the operations in the manufacture of these forgings are first described. After the projectile is rough forged the cavity is bored to size and the cylinder rough turned from the bourrelet to about 2 inches from the base concentric with each other leaving the

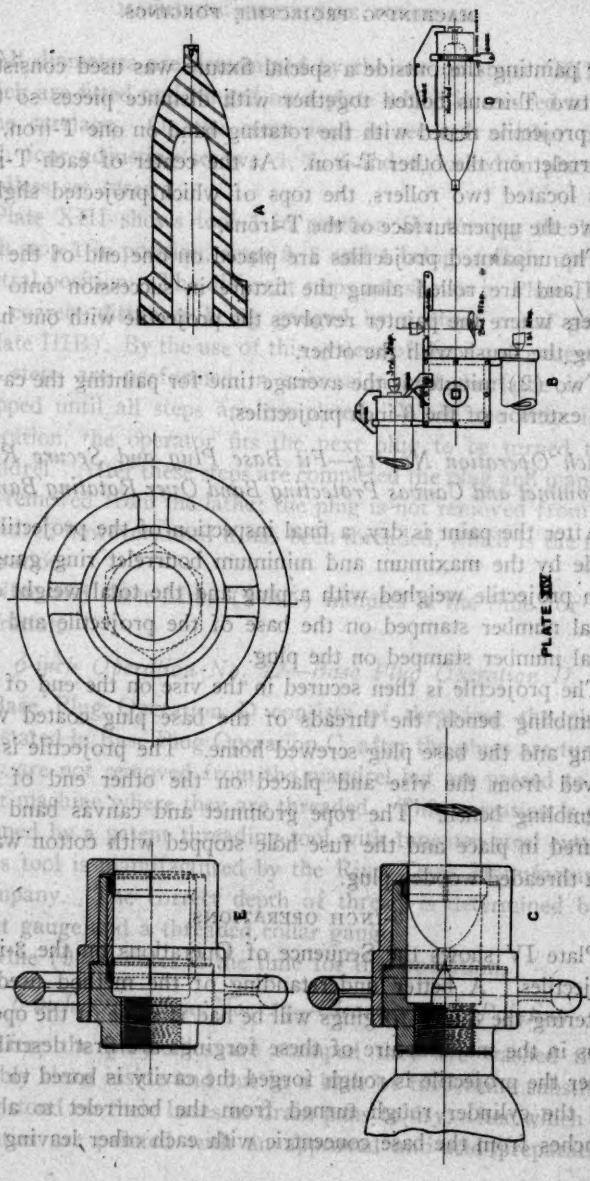


PLATE XIV

walls of equal thickness (Plate XIVA). The forging is then reheated and the rear end closed in; the dotted lines show the base of the forgings as received at Mare Island.

3-inch Operations Nos. 1 and 2—Centering.

The forgings are placed in the centering jig point down, a fixed center in the bottom of the jig entering the reamed center in the point of the forging. The chuck jaws are closed on the outside of the forgings by means of the crank; the yoke is brought into place and secured by a wing nut which forms a ball and socket joint in the yoke, thus insuring that the yoke itself is accurately in position. A center is then reamed in the base of the forging. This method insures the alignment of the centers in the point and the base of the forging. The forging is then removed from the jig and the center in the point re-reamed by a sensitive drill press to free it from the scale formed in heating the forging before closing in the base.

One (1) minute is the average time for one operator to perform both centering operations.

3-inch Operation No. 3—Rough Turn Body to 1-inch From Base (Photo 2).

In this operation the body is rough turned to 3 1/64 inches diameter to 1-inch from base. This work is done on centers using a common lathe dog as a driver. When the operator has once determined the rough diameter, the cutting tool is not drawn back from the work at the end of the cut. Instead, the feed nut is disengaged, the lathe stopped and the dead center released, the tool carriage being then returned to the starting point. By this method from ten to fifteen forgings are turned without readjusting the tool. When the operator has started a cut on a new forging he places the last forging turned in the fixture (2a) and removes the lathe dog. The forging is then reversed and placed in the fixture (2b) and the lathe dog placed on the point end of the forging making it ready for the 4th

operation on the next machine. The operator then places a dog on the next forging which he is to turn. It was found necessary for the operator on the first roughing cut to prepare the forging for the next operator for the reason that the next operator lost so much time changing driving dogs as to reduce his output and cause delay in all subsequent operations.

In this operation the speed of cut is 60 feet per minute, feed .0625-inch, depth of cut 15/64-inch.

Two and one seventh (2 1/7) minutes is the average time for this operation.

3-inch Operation No. 4—Complete Rough Turn Body, Rough Face Base, Rough Turn Rotating Band Groove.

This work is done on a Hendy Norton lathe equipped with a special revolving tool head, gauges, pointer and indexing dial. The revolving tool head is fitted with three cutting tools and an adjustable distance gauge. As previously stated, the operator in this operation neither secures nor removes the driving dogs, his time being devoted entirely to the machine work of this operation.

This operation consists of three distinct steps as shown in Plate XIVB. The first step removes the excess diameter at the base where the driving dog was secured in the 3d operation. This step requires no gauging, the cutting tool being brought up to the diameter turned in the 3d operation. The second step faces 5/16-inch off the base of the forging, a distance gauge being used to determine the amount of stock removed. This step determines the location of the cavity in the finished projectile and has a material bearing on the weight of the finished projectile.

The third step is rough turning the rotating band groove, the width of the cutting tool is 1/32-inch less than the minimum width of the finished groove. The distance from the base to the rear edge of groove is measured by an adjustable gauge, a sheet metal hand gauge being used for checking. The

depth of the band groove is determined by using the pointer and indexing dial. The use of this system and the gauges make it unnecessary to stop the lathe to gauge any dimension.

In this operation the speed of cut is 65 feet per minute, feed .0625-inch.

Two and one-fifth (2 1/5) minutes is the average time for this operation, complete.

3-inch Operation No. 5—Finish Turn Rotating Band Groove, Undercut Band Groove, Knurl Band Groove and Finish Turn Body.

This work is done on two Hendy Norton lathes fitted with revolving tool head, pot chuck, knurling device, dial and pointer. The combined tool head and knurling device is somewhat similar to that already described for the 6-inch projectiles.

Plate XIVC is a sectional drawing of the pot chuck and shows how the forging is gripped, the pot chuck, however, acts as a driver only, the forgings being turned on centers.

This operation consists of four distinct steps. The first step is to machine the rotating band groove to the depth and the minimum width; the depth is determined by using the dial and pointer, the width and distance from base of projectile is determined by the sheet metal gauge (XIVB).

The second step is to undercut the band groove. After the groove is undercut the tool head is drawn back until the slide reaches a stop when the center of the knurling roll is directly over and in line with the center of the lathe.

The third step is to knurl the band groove which is done by bringing the knurling roll in contact with the work by means of the crank and screw.

The fourth step is to finish turn the body. This step is finished in one cut, there being three diameters (in rear of band groove, band groove to bourrelet, and bourrelet), the allowed tolerances being respectively .006-inch, .010-inch and

.005-inch. These diameters are checked by maximum and minimum limit gauges. The distance from base to bourrelet is determined by the sheet metal gauge (XVID).

In this operation the turning speed is 60 feet per minute, feed .042-inch.

Three and one-half (3½) minutes is the average time for this operation, complete.

3-inch Operation No. 6—Finish Turn Point.

This work is done on two Hendy Norton lathes fitted with pot chucks and a combined steady rest and radius turning attachment. The forging is gripped in the pot chuck by two split rings, the rear interior face of the chuck acting as a stop (XIVE).

The base plate of the radius turning attachment was planed to fit the lathe ways and is held in place by two strong backs. This plate is fitted with a radius guide track in which works the radius turning arm which is pivoted at the correct distance from the lathe center to form the ogival of the desired radius. The radius turning arm is operated by a link attached to the lathe carriage, the screw feed being used. The points are finished in two cuts.

The cutting speed at the bourrelet is 125 feet per minute, diminishing as the tool travels toward the point. The feed at the bourrelet is .0234-inch and increases as the tool travels toward the point due to the link's decreasing its angle with the direction of travel of the carriage.

Four (4) minutes is the average time for this operation.

3-inch Operation No. 7—Banding.

The 3-inch projectiles are banded in the same press and in the same manner as described for the 6-inch projectiles, it being only necessary to change the segments at the end of each hydraulic cylinder. The jib crane and tongs are not necessary in handling the 3-inch projectiles; they are placed in and taken from the banding press by hand.

One 3-inch projectile was banded every 15 seconds. Two men are used, one handling the projectiles and the press, the other heating and placing the bands.

3-inch Operation No. 8—Drill Fuse Hole, Face Base and Round Edge, Ream and Counterbore Fuse Hole, Tap Fuse Hole, Rough Turn Rotating Band, Form and Size Rotating Band.

This work is done on a Jones and Lamson turret lathe. No special equipment is provided except taps, reamers, cutters, forming tools and tool holders.

The first step is to rough drill the fuse hole, face the base and round the periphery of the base. The drill is formed half round, the cutting edge being one-half the diameter of the drill; the radius of the body of the drill is also "backed off" to avoid interference if drilling an eccentric hole. A flat cutter is inserted in the tool holder, the end of the cutter entering a slot in the shank of the drill. As the drill nears its depth, the flat cutter faces the base and rounds the periphery of the base leaving a sharp corner at the fuse hole.

The second step is to ream the fuse hole to size and counterbore for the fuse. This is done by a reamer with a shoulder for forming the counterbore. The depth of the counterbore is determined by a stop. The fuse hole is tapped 18 threads per inch, left hand.

The rotating band is rough turned, then formed to size by a special forming tool, similar to that described in the 6-inch operations.

Two (2) minutes is the average time for this operation, complete.

3-inch Operation No. 9—Marking Base.

This work is done with the same air hammer as that used for the 6-inch, substituting a different anvil block and lettering die.

With this hammer, one man stamps 3-inch projectiles at the rate of 250 per hour.

3-inch Operation No. 10—Weighing and Stamping Weight on Base.

The projectiles are next weighed, the weight being stamped on the base in the space left blank for that purpose. The weight is marked in pounds and decimals. As the scales are graduated in pounds and ounces, a table converting ounces and fractions into decimals of a pound was printed and placed before the stamping bench.

3-inch Operation No. 11—Cleaning.

The projectiles are cleaned by immersion in a bath of turpentine substitute, after which they are drained base down.

3-inch Operation No. 12—Gauging, Inspecting and Shipping.

After the projectiles are dry, they are given a careful surface inspection, and are gauged by maximum and minimum ring gauges. They are then packed in shipping boxes having twelve separate compartments, the sides extending about 3 inches over the points of the projectiles, so that the boxes can stack, and provided with hand grabs for easy handling.

COAST GUARD CUTTERS

TAMPA, HAIDA, MODOC AND MOJAVE.

By Engineer-in-Chief Q. B. Newman, U. S. C. G., Member.

DESCRIPTION.

The Coast Guard Cutters *Tampa, Haida, Modoc and Mojave* are four similar vessels now building at the works of the Union Construction Company, Oakland, California. They are single screw vessels fitted with turbo-electric propelling machinery and designed for a speed of 16 knots at 1,620 tons trial displacement with the main motor developing 2,600 shaft horsepower. The *Tampa* is named for the Coast Guard Cutter *Tampa*, sunk with all hands in an encounter with a German submarine in Bristol Channel, September 26, 1918.

PRINCIPAL HULL DIMENSIONS.

Length over all, feet and inches	240–00
Length on water line, feet and inches	220–00
Beam, molded, feet and inches	39–00
Draft to L. W. L. mean, feet and inches	14–00
Displacement, 14-foot W. L., tons	1,640
Tons per inch immersion at L. W. L.	13.26
Block coefficient	.4765
Prismatic coefficient	.5328
Midship section coefficient	.8864
Area of midship section, square feet	482
Area of wetted surface, square feet	9,496

GENERAL REQUIREMENTS FOR MACHINERY.

In the design and construction of these ships, the nature of the duties they are to perform has been kept constantly in mind, and all selections have been made in conformity therewith.

The paramount duties of a Coast Guard Cutter, and the conditions imposed thereby, may be briefly summarized as follows:

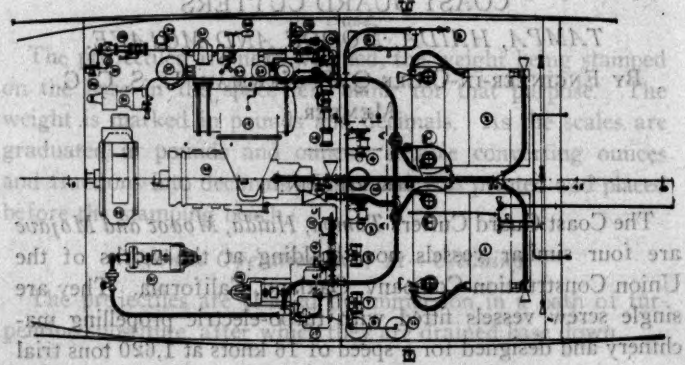


DIAGRAM OF MACHINERY.

(*a*) Rendering assistance to vessels in distress, and saving life and property. This calls for sea-worthiness and comfort in all weathers, speed, light draft and quick maneuvering for inshore work and towing ability.

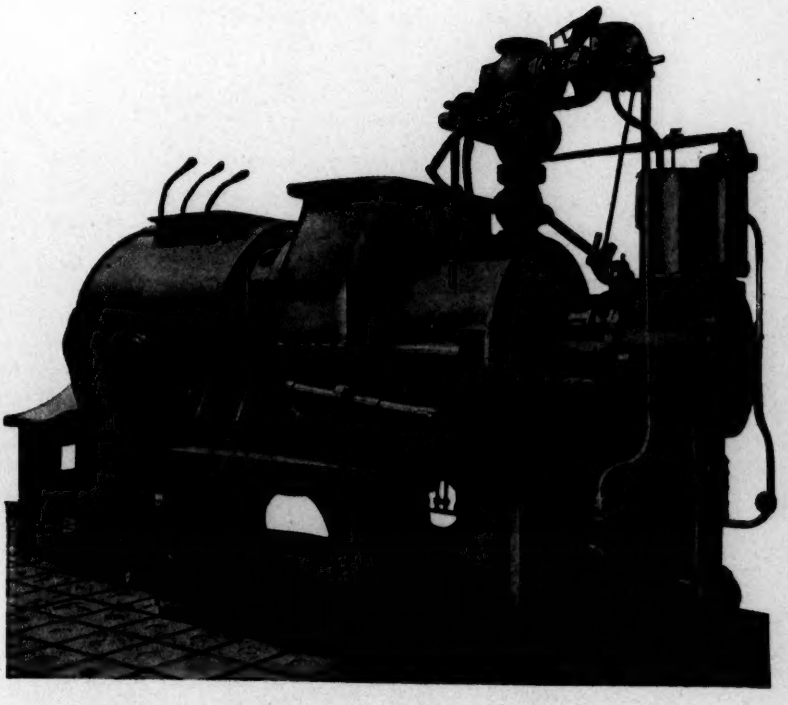
(*b*) Destruction or removal of wrecks, derelicts or other dangers to navigation. The requirements here are long legs and plenty of power.

(*c*) Protection of the seal in Alaska, extending medical aid to United States vessels engaged in deep-sea fisheries and observation of ice conditions in the North Atlantic during the spring season. For these duties all other considerations are subordinate to that of ability to keep the sea for long periods.

(*d*) Protection of the customs revenue. Same requirements as (*a*).

(*e*) Operating as part of the Navy in time of war. This need is met by the general characteristics of a gunboat.

In addition to the foregoing, prolonged absences from port and shop facilities demand machinery sufficiently rugged to



MAIN TURBO GENERATOR.

operate reliably for long periods, and of a type that may readily be repaired by the ship's force with the facilities on board.

With a statement of the conditions imposed the selection of propelling machinery becomes practically automatic. Efficiency, power, weight, space and cost select the steam turbine with water tube boilers and superheated steam; while responsiveness, comparative freedom from breakdown, and ease of effecting repairs dictate electric machinery for speed reduction between turbine and propeller.

PROPELLING MACHINERY.

The propelling machinery consists of a Curtis turbine driving an alternating current generator, which in turn supplies current to a synchronous motor on the propeller shaft. These three machines with appurtenances, and the exciters, are built by the General Electric Company, Schenectady, N. Y., and in design they conform to the standard practice of that Company. But an immense number of details had to be worked out to develop machinery of suitable size and power for these vessels, and particularly to adapt the synchronous motor for ship propulsion, which had not hitherto been attempted.

MAIN TURBINE.

The turbine is an eight-stage Curtis machine running at 3,000 revolutions per minute. It is designed to take steam at 200 pounds pressure (gauge) and 75 degrees superheat and exhaust against 2 inches (absolute) back pressure. Under these conditions, the guaranteed steam consumption is 11.4 pounds per hour per shaft horsepower delivered to propeller shaft. The first stage has two rows of blades and the other stages one row each.

Auxiliary exhaust steam in excess of feed heater requirements is admitted to the fifth stage of the main turbine. An automatic valve is installed to close this inlet in case the main turbine is stopped.

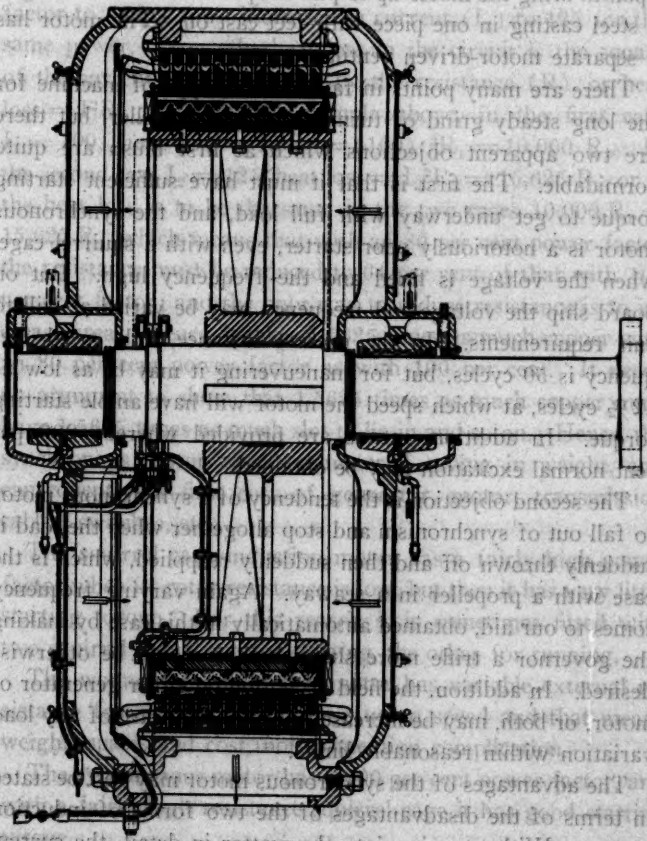
Speed is normally controlled by an oil governor which consists of a small gear type pump driven by the turbine and delivering oil beneath a piston working in a cylinder. The oil pressure varies with the turbine speed, and in case of increased pressure the piston rises in the cylinder and opens a pilot valve to a hydraulic cylinder, which in turn operates the controlling valve in the main steam line and reduces the turbine speed. Speed variations are obtained by means of a bypass in the pump discharge, and this bypass is directly operated by the speed lever on the operating panel. There is also an inertia governor which prevents speed exceeding 3,000 revolutions per minute. If the oil governor and the inertia governor should both be inoperative at the same time there is an overspeed tripping device which closes the main steam valve at 3,600 revolutions per minute. A fourth means of speed control is provided in an ordinary manually operated balanced throttle valve.

MAIN GENERATOR.

The turbine is direct connected to a two-pole, three phase, alternating current generator of the revolving field type. The output of the generator is 2,040 kilowatts at 2,300 volts. Ventilation is by means of fans incorporated in the rotor. The stator is built up of soft iron punchings assembled in a cast steel frame. The rotor is solid steel, forged integral with the shaft, the slots being milled in the forging. The rotor windings are thoroughly secured against motion in any direction by hardwood blocks, and protected against mechanical injury by heavy metal blocks and rings. All insulation is designed to resist the action of moisture and salt, while the permissible temperature rise is well beyond temperatures that are likely to be experienced.

MAIN MOTOR.

The propeller shaft is driven by a 46-pole, three phase, synchronous motor delivering 2,600 S.H.P. at 130 revolutions



MAIN MOTOR.

per minute. The bars of an ordinary squirrel cage induction motor are let into the pole faces, and this winding is depended upon to bring the motor up to pull-in speed. The stator ring is a steel casting in one piece with feet cast on. The motor has a separate motor-driven ventilating fan.

There are many points in favor of this type of machine for the long steady grind of turning a ship's propeller, but there are two apparent objections which at first blush are quite formidable. The first is that it must have sufficient starting torque to get underway with full load, and the synchronous motor is a notoriously poor starter, even with a squirrel cage, when the voltage is fixed and the frequency high. But on board ship the voltage and frequency may be varied at will to suit requirements. In the present instance, full speed frequency is 50 cycles, but for maneuvering it may be as low as 12½ cycles, at which speed the motor will have ample starting torque. In addition, means are provided whereby 200 per cent normal excitation may be obtained.

The second objection is the tendency of a synchronous motor to fall out of synchronism and stop altogether when the load is suddenly thrown off and then suddenly reapplied, which is the case with a propeller in a seaway. Again varying frequency comes to our aid, obtained automatically in this case by making the governor a trifle more sluggish than would be otherwise desired. In addition, the field excitation on either generator or motor, or both, may be increased so as to take care of any load variation within reasonable limits.

The advantages of the synchronous motor may best be stated in terms of the disadvantages of the two forms of induction motors. Without going into the matter in detail, the current in an induction motor is not in phase with the voltage, but lags behind it by a certain angle. The cosine of this angle is called the power factor. The power (P) of a three phase motor is $\sqrt{3}$ times the product of volts (E), ampères (I), and the power factor (P.F.). That is, $P = \sqrt{3} \times E \times I \times P.F.$

Hence, for a given power the current is inversely proportional to the power factor. Suppose for example that when the power factor is 1.00, the current (I_1) is 100; if we reduce the power factor to .80 we must increase the current (I_2) to 125 for the same power. Now, the heat loss in the circuit is the square of the current (I^2) multiplied by the resistance (R), or heat loss $= I^2R$. Taking the example above, in the first case $I_1 = 100$, heat loss $= I_1^2R_1 = (100)^2R_1 = 10{,}000\ R_1$. In the second case $I_2 = 125$, heat loss $= I_2^2R_2 = 15{,}625\ R_2$, or if the heat loss is to be the same in the two cases $10{,}000\ R_1 = 15{,}625\ R_2$, which means that with an 80 per cent power factor the resistance must be reduced to 64 per cent of that with 100 per cent factor; and the only way to reduce resistance is to increase area. Hence, it takes 1.5625 times as much copper with an 80 per cent power factor as with 100 per cent. It needs no argument to show that 1.5625 times as much copper must have 1.5625 times as much slot to lie in and so on. Hence, the synchronous motor effects a material saving in weight, and consequently in first cost of generator, motor, transmission cables, switches, etc.

The squirrel cage induction motor has a fairly high power factor when its rotor resistance is low, but then it has very little starting torque. For this reason it is sometimes fitted with two squirrel cages, one for starting, the other for running.

The wound rotor induction motor has variable external resistance for starting and coming up to speed and that means weight, space, and cost, not to mention complication.

The synchronous motor has a 100 per cent power factor, and with a fairly high resistance squirrel cage it has good starting torque.

The chief objection to the synchronous motor for a marine installation is the relatively large diameter of the stator ring, the diameter of this present machine being about 10 feet 6 inches. Even this is smaller in diameter than it would be if

built for shore installation where space occupied is not a prime consideration. The length of the motor is somewhat increased to make up for decreased diameter.

But it is not alone in the saving in weight and cost and the increased efficiency that the synchronous motor is superior. From the operating point of view it has very real advantages. Momentary double excitation of generator and motor fields gives it a responsiveness in maneuvering not possessed by any other form of ship propulsion apparatus and this is of the highest importance for a vessel operating, as Coast Guard vessels so frequently do, in close proximity to reefs and shoals.

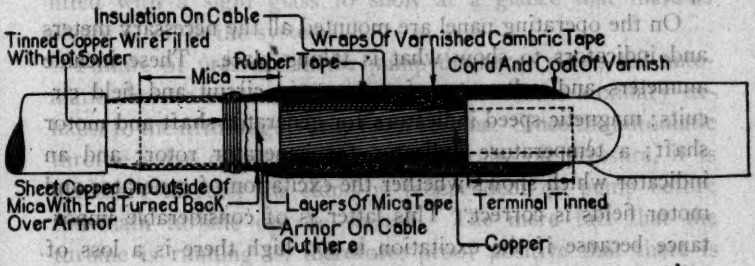
An induction motor designer, jealous always of his power factor, must keep the width of air gap down to the very minimum. Consequently, before stator coils can be removed or repaired, the rotor must first be removed, which requires lifting gear more robust than is generally carried on board ship. The air gap on a synchronous motor is generally wide, and repairs to stator coils can be made without disturbing the rotor. In addition, the squirrel cage is put on in sections which are bolted together and the field poles are bolted to the rim of the rotor, so that the D.C. field may be removed from the rotor entirely, affording complete access to every part of the machine without lifting heavy weights.

CONTROLS.

There are two groups of contactors, or switches, one for the fields and one for the main circuits. Normally, each contactor is operated by a solenoid, but in case of failure of the solenoids there is a cam shaft for manual operation, so arranged that throwing a lever on the cam shaft produces the proper sequence of opening and closing of contactors. All contactors are of the magnetic blowout type with arc chutes. Under normal conditions, however, the high tension cables will always be dead before the main contactors are opened. There are two control levers mounted on the operating panel,

one for speed and the other for direction. These are interlocked so that the motor cannot be reversed until the turbine speed lever is moved to "SLOW."

The speed lever, as already stated, opens a bypass in the oil governor line to increase the speed of the turbine. The reversing lever is geared to a drum, similar to a street car con-


Insulation On Cable
Tinned Copper Wire Filled With Hot Solder
Wraps Of Varnished Cambric Tape
Rubber Tape
Cord And Coat Of Varnish
Mica
Sheet Copper On Outside Of Mica With End Turned Back Over Armor
Layers Of Mica Tape
Terminal Tinned Copper
Armor On Cable Cut Here

MAIN CABLE.

troller, with sectors for operating the solenoids on the main and field contactors. When a signal is received to reverse, the operator slows the turbine to quarter speed, brings the reversing lever slowly through neutral to reverse, and then brings the turbine up to the desired speed. While it is not at all necessary that the operator should know what is happening, the sequence of events is as follows: (1) The generator field is killed and the motor becomes a generator, expending the energy of its momentum and the turbine effects of the propeller in sending current through the short circuit of the generator armature, this brakes the motor to rest; (2) the motor field is killed; this leaves the high tension circuit dead; (3) and (4) two phases of the main circuit are reversed which reverses the direction of phase rotation on the motor; (5) the generator field is connected; this generates current in the main circuit and brings the motor up to speed as an induction motor; (6) the motor field is connected and the motor pulls into step, after which the speed is increased to whatever is desired.

For ordinary "backing and filling," as in coming to anchor or getting alongside or away from a wharf, it will not be necessary to put on the motor field at all, the squirrel cage winding being ample for this purpose. In fact, if the motor field should be entirely disabled at sea the motor could bring the ship to port operating as an induction motor at reduced speed.

On the operating panel are mounted all the necessary meters and indicators to show what is taking place. These include ammeters and volt-meters for the main circuit and field circuits; magnetic speed indicators for generator shaft and motor shaft; a temperature indicator for generator rotor; and an indicator which shows whether the excitation of generator and motor fields is correct. This latter is of considerable importance because if the excitation is too high there is a loss of efficiency, while if it is too low there is danger that the motor may fall out of step, particularly if the propeller tends to race. Resistance grids in the field circuits furnish the means for varying excitation to suit the requirements. The total weight of all contactors and control devices including the operating panel is only 6,500 pounds.

WEIGHTS OF PROPELLING MACHINERY.

The installed weights are as follows:

Turbine, pounds	28,000
Generator, pounds	33,000
Motor, pounds	57,429
Control group, pounds	4,105
Control panel, pounds	2,313
Total weight of propelling machinery and controls, pounds	124,849
Weight per S. H. P., pounds	48

LUBRICATION SYSTEM.

Lubrication requirements are extremely modest, forced lubrication being limited to the straight cylindrical bearings of the turbo-generator and the motor, the thrust bearing of the turbine

and the main thrust bearing. Oil for these parts is furnished by the same gear pump which operates the oil governor or by an independent reciprocating steam pump, if desired. The oil pumps take oil from the sump tanks and deliver it through the coolers into the overhead tanks, from which it flows, by gravity, through the bearings. The overflow from the gravity tanks is fitted with a sight glass to show at a glance that there is plenty of oil in the system.

Failure of pressure in the pump discharge closes the automatic throttle valve and the valve which admits auxiliary exhaust to the fifth stage of the main turbine. Since the automatic throttle valve cannot be opened except by oil pressure, it is necessary to start the stand-by independent oil pump before the main turbine can be started. The mere fact that the turbine is running is, therefore, proof positive that there is pressure in the pump discharge. Each principal bearing is fitted with a sight feed and a thermometer.

There are two oil coolers and an oil cooler circulating pump. Normally, the oil coolers take their water from the main circulating pump.

Two gravity tanks, located in the engine room trunk, have a capacity of 200 gallons each. Two sump tanks with 400 gallons total capacity are installed under the turbo-generator.

Two reserve tanks, each with a capacity of 400 gallons, are located aft, close up under the berth deck, high enough to drain into the sump tanks.

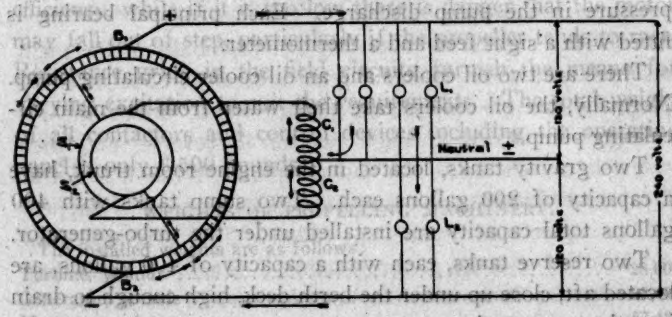
All tanks have filling pipes from the weather deck.

There is one centrifugal oil purifier for removing water and other foreign matter from the oil. Piping is arranged so that oil from a gravity tank can be cleaned in the purifier, drained into the sump tanks and returned to its own tank after the latter has been cleaned. This permits the thorough cleaning of the entire system without loss of oil.

EXCITERS.

There are two 240-volt, 75-kilowatt exciters, either of which is adequate for exciting the main generator and motor and for furnishing current for ship's lighting. Each machine is driven by a Curtis turbine through a Poole planetary gear with a three-to-one reduction.

The exciters are three-wire machines, and the wiring is so arranged that either full or half voltage is available as desired. For example, it is necessary in maneuvering to have momentary over-excitation of the fields, and to obtain this the full voltage is used. At other times the generator field is on one 120-volt leg and the motor field on the other, giving practically a balanced load on the exciter. The ship's lighting circuits, shop motors, etc., are also arranged for a balanced load.



3-Wire Generator and External Circuits.

Devices are installed on the exciter switchboard for equalizing and paralleling the two machines. It is not intended to operate both of them at once, but without the equalizer it would be necessary to stop the propelling machinery in order to shift to the second exciter were the first to become disabled. It should be stated that an equalizer for three-wire generators is a much more complicated device than for two-wire machines and its manipulation by a careless operator is likely to be at-

tended with interesting circumstances. There is no reason, however, why a properly instructed person should not operate it with entire success.

The principle of the three-wire generator is briefly outlined here for the benefit of those who may not be entirely familiar with it.

The voltage between positive and negative brushes B_1 and B_2 is 240 volts; and the voltage between the neutral wire and either of the others is half the total, or 120 volts. If the loads L_1 and L_2 on the two legs are exactly equal there will be no current in the neutral wire. But if L_1 is greater than L_2 then the neutral wire must carry the difference $(L_1—L_2)$. The neutral wire is connected to the middle point of a "compensator," which is a coil of heavy copper wire wound on a soft iron core (C_1, C_2), and connected through slip rings S_1 and S_2 to two commutator bars. It is readily seen that current flowing in on the neutral wire divides into two equal parts, each part going through half of the compensator, through the connection to the brush and slip ring, to one commutator bar. Here each part divides again, not equally this time though, and flows to the positive and negative brushes B_1 and B_2.

The compensator is the interesting feature. Its ohmic resistance must be low or else an undue current would be forced through L_2. But it is short circuited across the commutator and so, to prevent a heavy short circuited current, it is given a very high reactance, because the current tending to flow through it is alternating, and hence can be held back by high reactance.

In practice this current is limited to only a very small charging current. The direct current $(L_1—L_2)$ is not affected by the reactance in the compensator. The neutral wire and the compensator are designed to stand complete unbalance, that is, to carry a full load on one leg and no load on the other, although, as stated elsewhere, the two loads are as nearly equal as practicable.

Each exciter set weighs 8,400 pounds, and the exciter switchboard 1,382 pounds.

SHAFTING AND BEARINGS.

Turbine Shaft.

Diameter, maximum	9 inches.
Length over all	7 feet, 6⅛ inches.
Forward journal	6 inches × 6 inches.
After journal	9 inches × 10 inches.

Generator Shaft.

Diameter, maximum	14⅝ inches.
Length over all	7 feet, 3⅞ inches.
After journal	8 inches × 8 inches.

Motor Shaft.

Diameter, maximum	15 inches.
Length over all	8 feet 4 inches.
Forward journal	12 inches × 12 inches.
After journal	12 inches × 12 inches.

Thrust Shaft.

Diameter	12 inches.
Length	16 feet 0 inches.

Intermediate Shaft.

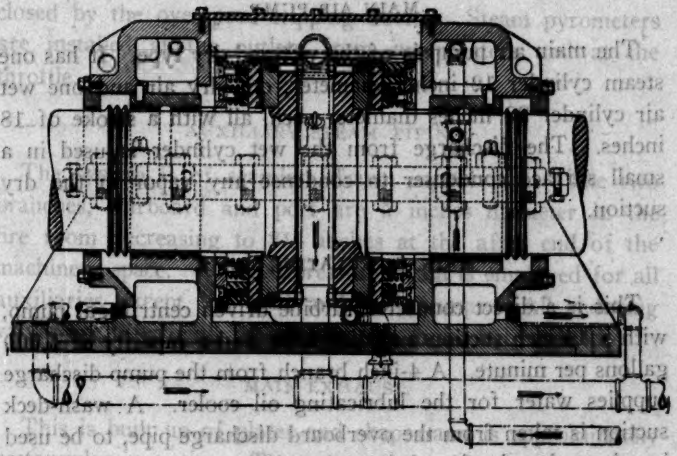
Diameter	12 inches.
Length	24 feet 0 inches.

Propeller Shaft.

Diameter	13 inches.
Length	29 feet 0 inches.

THRUST BEARING.

The main thrust bearing is on the propeller shaft abaft the main motor and on a separate foundation. It is of the spring type made by the General Electric Company. There is one collar on the shaft which transmits the thrust to a flexible plate backed up by a large number of helical springs. In case there is a tendency to undue pressure at any point the flexible



THRUST BEARING.

plate yields at that point and thus maintains a uniform pressure over its entire surface. The housing is entirely full of oil circulated by the main lubrication system.

PROPELLER.

There is one four-bladed, right hand, solid, cast steel propeller of 13 feet diameter, 14 feet pitch and having a helicoidal area of 63.11 square feet.

MAIN CONDENSER.

The main condenser has a cast iron cylindrical shell, 58 inches diameter by 9 feet long. There are 2,444 tubes, 5/8 inch outside diameter, with a cooling surface of 3,600 square feet. Openings are provided for an 8-inch wet and a 6-inch dry air pump suction. The main exhaust inlet is rectangular, 24 inches high by 7 feet long. The tubes are arranged to facilitate rapid diffusion of steam.

MAIN AIR PUMP.

The main air pump is of the wet-and-dry type. It has one steam cylinder 12 inches diameter, one dry air and one wet air cylinder 22 inches diameter each, all with a stroke of 18 inches. The discharge from the wet cylinder is used in a small surface condenser to condense any vapor in the dry suction.

MAIN CIRCULATING PUMP.

This is a direct connected turbine driven centrifugal pump, with a 14-inch suction and discharge, and a capacity of 5,000 gallons per minute. A 4-inch branch from the pump discharge supplies water for the lubricating oil cooler. A wash-deck suction is taken from the overboard discharge pipe, to be used in winter for clearing the weather decks of snow and ice.

BOILERS.

There are two Babcock & Wilcox straight tube, water tube, oil-burning boilers installed in a watertight compartment. Each boiler has 2,900 square feet of heating surface and 290 square feet of superheating surface. The superheaters are between the first and second passes with tubes athwartship. The main and auxiliary steam connections and the safety valves all have separate mountings on the drum. Each boiler has four burners and pressure atomization is employed.

MAIN STEAM PIPING.

This is wrought steel with cast steel valves and fittings. The main pipe is 6 inches with a 4½-inch branch to each boiler. Connections are made for using saturated steam in the main turbine in case superheaters are disabled. A monel metal strainer is inserted above the throttle. There is an hydraulically operated valve for automatic speed regulation, a manually operated balanced throttle valve, and a clapper valve which is

closed by the overspeed tripping device. Steam pyrometers are installed at the outlets from superheaters and at the throttle.

AUXILIARY STEAM PIPING.

The system is of seamless drawn copper pipe. The two branches, starboard and port are 5 inches diameter in the fire room decreasing to 2½ inches at the after end of the machinery space. Reduced pressure steam is employed for all auxiliaries except the feed and fuel pumps. All reducing valves are fitted with pressure gauges and are by-passed.

MAIN EXHAUST.

This is built up of plates and shapes and is approximately rectangular in section. The area at the turbine nozzle is about 9¾ square feet, and at the condenser shell about 18½ square feet. No provision is made for expansion of this pipe.

AUXILIARY CONDENSING PLANT.

This is the conventional unit plan, consisting of a 600-square foot condenser mounted directly on a reciprocating air and circulating pump. The condenser has a cylindrical cast iron shell of 25 inches inside diameter and 7 feet long between tube sheets. There are 521 tubes ⅝-inch outside diameter by 7 feet 2 inches long. The combined air and circulating pump has 7½-inch steam, 10-inch water, and 10-inch air cylinder, all with a stroke of 10 inches.

MAIN FEED PUMP.

This is a vertical, simplex pump, with a 10-inch steam cylinder, 6-inch water cylinder, and a stroke of 24 inches. It is piped to take fresh water only and discharge to the main feed line only. It is installed in the fire room.

AUXILIARY FEED PUMP.

This is a duplicate of the main feed pump and is also installed in the fire room. It has suctions from the fresh water tanks, the fresh water side of the main condenser, the hot salt water side of the main condenser, the sea and the bilge and discharges to the auxiliary feed line, the feed tank, the reserve tank, the fire main and overboard.

FEED WATER HEATER.

This is a two-pass, coil type heater of sufficient capacity to heat 35,000 pounds of water per hour from 90 degrees to 220 degrees F., with 10 pounds (gauge) pressure on the steam side. It is piped to the main feed line only.

FUEL OIL SYSTEM.

There are four burners on each boiler. They were made and furnished with the boilers by the Babcock & Wilcox Company. They are designed for operation under either forced or natural draft, requiring 2½ inches pressure for full power.

There are two oil heaters, installed so that either, neither, or both may be used. Each is capable of heating 3,000 pounds per hour of oil of 16 degrees gravity from 60 degrees to 200 degrees (F.) with 80 pounds steam pressure. There is one 1½-inch oil meter calibrated to read in gallons and correct for oil having a specific gravity of 0.8981.

There are two vertical duplex fuel oil pumps, the cylinder diameters being, steam 5¼ inches, oil 3½ inches and stroke being 5 inches. High and low tank suctions are provided and the discharges are to the burner line, or overboard or to the tank filling line either direct or from the dead end of the burner line. Suitable arrangements are made of pipes and manifolds so that either pump may be used for any purpose while the other pump is being used for any other.

FORCED-DRAFT BLOWER.

There is one forced-draft blower located in, and taking air from, the engine room, air being exhausted direct into the closed fire room. The blower is mounted on the berth deck level in the starboard forward corner of the engine room. The fan is 22½ inches diameter and has a capacity of 18,000 cubic feet per minute against 2½-inch head. It is direct connected to a 24-inch Terry turbine.

AUXILIARY EXHAUST.

The exhaust system has been designed with special attention to normal underway operating conditions. Such exhaust steam as is not condensed in the feed water heater is admitted to the fifth stage of the main turbine, so that its remaining available energy is extracted in the last four stages. This permits the carrying, without loss, of a back pressure of 8 to 10 pounds (gauge) on all auxiliaries for feed heating and the necessary sizes of pipes and fittings are also much reduced. The inlet to the fifth stage is 6 inches in diameter, which just accommodates the maximum auxiliary exhaust at 8 pounds pressure and obviates the necessity of a back pressure regulating valve in the system. A manually operated stop valve in the fifth stage line can be throttled to keep up the back pressure at lower powers. A combined stop-relief valve in the auxiliary exhaust to main condenser furnishes the other necessary element of control, and at the same time prevents excessive pressures in the system. Connection is made to the auxiliary condenser for port use and there is an atmospheric exhaust. The only possibilities of air leakage into the main condenser are in the main turbine and main exhaust.

REFRIGERATOR PLANT.

There is a one-ton refrigerating unit of the Audiffren-Singrun type. It is driven by a 5 horsepower direct current

motor. Connections are made to the refrigerating space and to the scuttle butt.

MACHINE SHOP.

The shop is located on a flat at the berth deck level in the starboard after corner of the engine room. In addition to a complete outfit of hand tools there are a 14-inch by 6-foot lathe, a 20-inch drill press, and a grinder with two wheels. Each machine has a separate motor drive.

STEERING GEAR.

The steering gear is of the hydro-electric type, consisting of a double ram working in two cylinders and direct connected to a slotted tiller. Pressure for the ram is supplied by a Waterbury variable stroke pump running at 600 revolutions per minute. Power is furnished by a 6-horsepower, constant speed motor. Control of the gear is by means of a telemotor, or manual as desired. Two emergency hand steering wheels, located aft, are geared to the tiller.

ANCHOR WINDLASS.

This is a steam windlass with two horizontal wild cats and two capstans. It is mounted on the upper deck. The wild-cat shafts are vertical. The engine, located on the berth deck, is a two-cylinder reversible engine with 8-inch by 8-inch cylinders. Control is from either the upper deck or the berth deck.

CAPSTAN.

The after capstan is located on the port side of the upper deck in a convenient position for warping and for handling towing lines. It is connected through a pair of gears and a worm and wheel to a 20-horsepower electric motor on the berth deck.

THE CRITICAL SPEED OF PROPELLER SHAFTS.

By Prof. P. E. Brunelli.*

Propeller shafts of light ships are often liable to work very near their critical speed. In many cases, as I have been able to verify, the safety margin is much more narrow than one likes to think, and there are on record cases where this margin has been entirely lacking, causing heavy losses, financial and otherwise. Therefore the matter is worthy of careful consideration, and I think a simple formula to solve the problem will be welcome.

The formula I propose is the following:

$$\omega_c = \left(1 - 3.3\,\frac{\lambda^3}{\varphi^4}\right)\omega_0$$

$$\omega_0 = \frac{\pi^2}{l_1^2}\sqrt{\frac{gEI}{p_1}}$$

Here:

ω_c = critical speed of the propeller shaft (rad. per sec.). To obtain the value in revolutions per minute multiply by 9.55.

ω_0 = value of the critical speed considering only the span between the two last bearings and assuming these to be simply supported.

g = gravity acceleration.

E = Young's modulus.

I = moment of inertia of the section of the shaft.

λ, φ = numerical coefficients which are obtained as follows.

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Call:

l_1 = the distance between the centers of the last two bearings.

l_2 = the distance between the center of the aft bearing and the end of the hub.

Then:

$$\lambda = l_2 / l_1$$

Again call:

p_1 the weight of the shaft per unit length.

l the length of the hub.

Q the weight of the propeller.

Q^1 the weight of the shaft length in the hub of the propeller

and let $$p_2 = \frac{Q+Q^1}{l}$$

Then:

$$\varphi^4 = p_1 / p_2$$

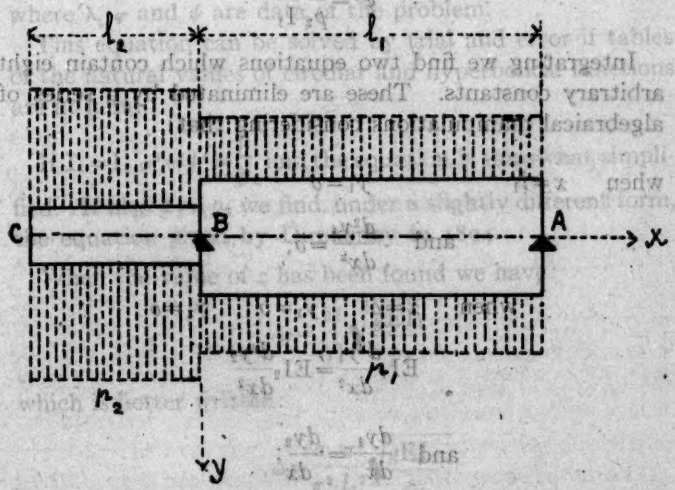
Take any units provided they are all homogeneous; i. e. if g is measured in feet per second per second all other lengths must be expressed in feet; E in pounds per square foot; p in pounds per foot run and so on.

The formula is an empirical one. I am going to show presently that within the limits of the usual applications, its results are very nearly concordant with those obtained by exact mathematical treatment.

I must observe here that the particular case where $\varphi = 1$ (a case which has nothing to do with propeller shafts) was treated many years ago by Professor Dunkerley and Professor Chree. Professor Chree gave also an approximate formula with which, for this particular value of φ, the formula given above is exactly coincident.

The value of the critical speed for complicated cases of shafts, when load and diameter are far from uniform cannot be obtained mathematically, and even for the simpler cases the solution of the problem is not very handy. On the other hand approximate graphical solutions do not lend themselves to the discussion of the problem and cannot be epitomized in a single formula. The nearest approach to the conditions of a propeller shaft that I can find to be amenable to calculation is the following. We shall discuss later its approximation.

Let us consider a shaft supported at two points, one of them at one end, the other intermediate; we have then two spans. Each span is of constant section and uniformly loaded, but section and load can differ in the two spans. Let us call l_1, l_2 the lengths of the two spans; p_1, p_2 the corresponding loads per unit length; I_1, I_2 the moment of



inertia of the two sections. We shall refer the system to a couple of coordinate axes with the origin in the intermediate

support as shown in the annexed sketch. Then by the usual theory of the whirling speeds, the deflections in the two spans are given by:

$$\frac{d^4y_1}{dx^4}=m_1^4y_1 \text{ and } \frac{d^4y_2}{dx^4}=m_2^4y_2$$

if we substitute:

$$m_1^4=\frac{p_1}{I_1}\frac{\omega^2}{gE} \text{ and } m_2^4=\frac{p_2}{I_2}\frac{\omega^2}{gE}$$

then we have

$$\frac{m_1}{m_2}=\varphi$$

if

$$\varphi^4=\frac{p_1}{p_2}\frac{I_2}{I_1}$$

Integrating we find two equations which contain eight arbitrary constants. These are eliminated by a series of algebraical manipulations considering that:

when $x=l_1$ $y_1=o$

and $\frac{d^2y_1}{dx^2}=o$;

when $x=o$ $y_1=o$ $y_2=o$

$$EI_1\frac{d^2y_1}{dx^2}=EI_2\frac{d^2y_2}{dx^2}$$

and $\frac{dy_1}{dx}=\frac{dy_2}{dx}$;

and last when $x=-l_2$

$$\frac{d^2y_2}{dx^2}=o \qquad \frac{d^3y_2}{dx^3}=o$$

Putting again for brevity's sake

$$\psi^4=\frac{p_1I_1}{p_2I_2}$$

and

$$m_1l_1=z$$

the final equation is:

$$(\cot z-\coth z)\left(\tanh\frac{\lambda}{\varphi}z-\tan\frac{\lambda}{\varphi}z\right)=\frac{2\psi^2}{\varphi}$$

$$\left(1+\frac{1}{\cos\frac{\lambda}{\varphi}z\cosh\frac{\lambda}{\varphi}z}\right)$$

where λ, φ and ψ are data of the problem.

This equation can be solved by trial and error if tables of the natural values of circular and hyperbolical functions are at hand.

If $I_1=I_2, \varphi^4=\psi^4=\frac{p_1}{p_2}$ and the equation is somewhat simplified. If also $p_1=p_2$ we find, under a slightly different form, the equation given by Dunkerley in 1894.

When the value of z has been found we have:

$$\omega=\frac{z^2}{l_1^2}\sqrt{\frac{gEI_1}{p_1}}$$

which is better written:

$$\omega=\frac{z}{\pi^2}\frac{\pi^2}{l_1^2}\sqrt{\frac{gEI_1}{p_1}}$$

$$=\frac{z^2}{\pi^2}\omega_0$$

$$= 0.10132^2\omega_0$$
$$= \beta\omega_0$$

where the meaning of ω_0 has been explained above.

I have solved the equation for a series of values of λ and φ when $I_1 = I_2$ and the values of β which result are given in the following table:

$\frac{l_2}{l_1}$	$\frac{1}{\varphi}$ = 1.0	1.5	2.0	2.5
	β			
0.05	0.9996	0.9979	0.9932	0.9840
0.10	0.9968	0.9838	0.9440	0.8759
0.15	0.9888	0.9438	0.8314	0.6639
0.20	0.9736	0.8689	0.6655	0.4733
0.25	0.9480	0.7607	0.5140	0.3464

The values of β as given by the empirical formula stated above are shown in the following table with their differences from the true value.

$\frac{l_2}{l_1}$	$\frac{1}{\varphi}$ = 1.0		1.5		2.0		2.5	
	β	diff.	β	diff.	β	diff.	β	diff.
0.05	0.9996	0.0000	0.9979	0.0000	0.9934	−0.0002	0.9839	+0.0001
0.10	0.9967	+0.0001	0.9833	0.0000	0.9472	−0.0032	0.8711	+0.0048
0.15	0.9889	−0.0001	0.9436	+0.0002	0.8218	+0.0096	0.5649	+0.0990
0.20	0.9736	0.0000	0.8663	+0.0026	0.5776	+0.0879	−0.0312	
0.25	0.9489	−0.0004	0.7390	+0.0217	0.1750	+0.3390	−1.0141	

The table shows that so long β is greater than 0.75 or thereabouts the simple empirical formula is very reliable and approximate.

We must now consider if the supposed conditions can be said to apply to a propeller shaft. Of course there are differences, but they are of the same order as must too often be allowed in engineering work.

First, the shaft is not supported at points but has long bearings. Of course this cannot be accounted for in ordinary calculations. Some might perhaps be tempted to consider the shaft as *encastré* but with the values of the play in such bearings this is absolutely out of the question. It can be verified easily that the ordinary values of the length of the bearings and of the play allow a larger value of the inclination of the elastica than could be realized in a shaft simply supported, and this disposes of any idea of *encastrement*.

Second, the two spans we have considered are not the whole shaft which is continued towards the engine. If the next span is short, its presence can raise the value of the critical speed by something like 20 or 30 per cent. If it is long there is no appreciable gain in the value of the critical speed.

Third, there are variations in the diameter of the shaft. Slight differences where the shaft is supported by the bearings are not of great consequence. But we have neglected the tapering of the shaft end and this is not without influence, even if, in hollow shafts it is partially compensated for by a reduction of the bore. On the other hand, the shaft is enormously stiffened by the hub of the screw and there is no doubt that this more than compensates for the effect of the tapering of the shaft.

Fourth, the screw is not exactly a load uniformly distributed.

Fifth, the righting effect of the transverse dimensions of the screw has not been taken into account.

Sixth, the load of the screw is not extended to the point where the shaft is supposed to be supported. The method

we have indicated for covering this point of the calculation tends to minimize any effect of this discrepancy.

Seventh, we have not taken into account the effect of the thrust of the screw, and this can probably lower the critical speed by something like 2 or 3 per cent.

In a few words the main facts neglected by the formula all tend to raise the critical speed above the calculated value, and as item 7 (which tends to lower it) is not of a great amount the calculated value allows a certain margin of safety. All these items however are not enough to explain the satisfactory behavior of certain shafts, which sometimes are run at speeds which exceed the calculated value by more than these same items can justify. The redeeming feature must be therefore, the damping effect exerted by the water in which the shafts revolve. This however shows how easy it is to approach very near to a perilous condition with installations which are meant to have a large factor of safety.

The following example shows the method of calculating critical speed by the formulae already given.

NUMERICAL EXAMPLE.

(*metrical units*).

External diameter	meters	0.125
Inner diam.	meters	0.065
Load per meter (p_1)	Kg. per meter	69.7
Moment of inertia (I)	(meters)^4	11.108×10^{-8}
Young's modulus (E)	Kg. per sq. m.	2.2×10^{10}
$E \times I \times g$		239.5×10^4
$E \times I \times g \div p_1$		3.4360×10^4
$\sqrt{E \times I \times g \div p_1}$		185.5
$l_1 = 0.274 + 5.610 + 0.215$	meters	6.099
$\omega_0 = \frac{\pi^2}{l_1^2}\sqrt{EIg \div p_1}$	rad. per sec.	49.20

ω_0	rev. per min.	470
Length of the hub	meters	0.264
Weight of the screw	Kg;	254
Weight of the screw per meter	Kg. per met.	962
Load on the external span (p_2)	Kg. per met.	1032
$p_2 \div p_1\ (1/\varphi^4)$	num.	14.8
$l_2 = 0.264 + 0.274$	meters	0.538
$l_2 \div l_1\ (\lambda)$	num.	0.0882
$\lambda^3 \div \varphi^4$	num.	0.01015
$1 - 3.3\lambda^3 \div \varphi^4$	num.	0.9665
Critical speed (ω)	rev. per min.	454

A PRACTICAL METHOD OF CALCULATING THE POWDER PRESSURE AND THE VELOCITY OF PROJECTILE ALONG THE BORE OF A CANNON.

By ALBERT E. GUY, Major, Ord. Dept., U. S. A.,
ASSOCIATE MEMBER.

The Leduc ballistic formulas are in current use in France and have been adopted by the U. S. Army and Navy Ordnance—with suitable modifications to meet powder characteristics—because their application has furnished results consistent with practice. The following is a mere development of these formulas intended to facilitate their application to drafting room work.

Charts I and II are used for calculating and delineating the powder pressure curve; Chart III is used for establishing the velocity curve of the projectile throughout the bore of the gun.

From the point of view of the delineator the design of a gun is based upon the following factors:

C, The caliber or inside diameter of the bore as measured across two opposite lands; it is expressed in inches.

u_o, the length of travel of the rear of the projectile along the bore, in inches.

P_p, the maximum effective pressure on the base of the projectile, in pounds per square inch.

P_b, the maximum pressure exerted by the powder gases on the breech, in pounds per square inch.

p, the weight of the projectile, in pounds.

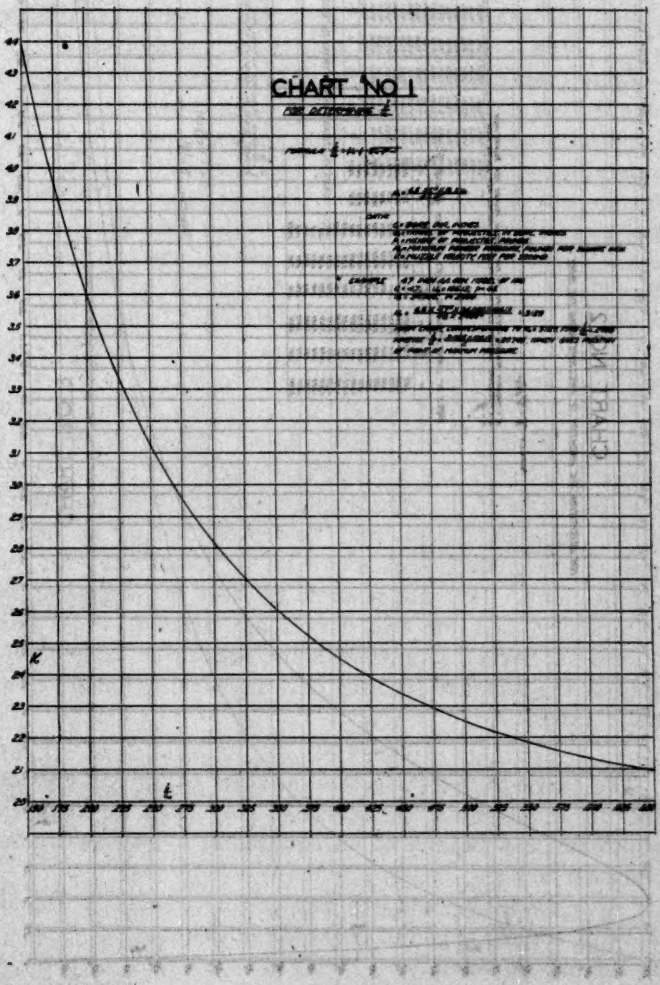


CHART NO. 1
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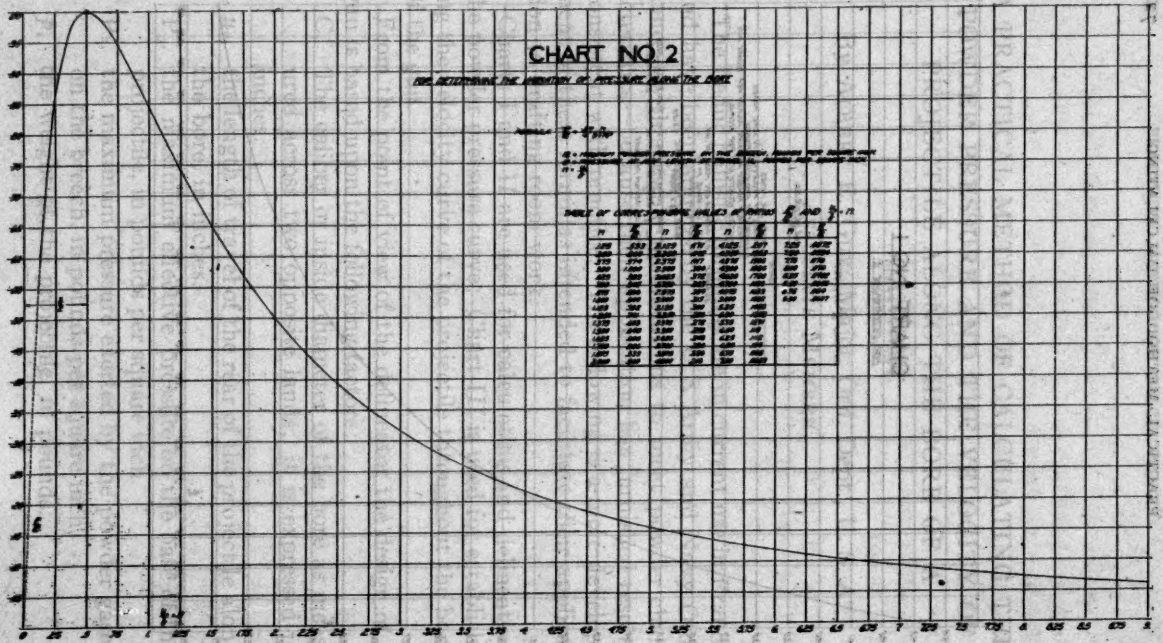
CHART NO. 2

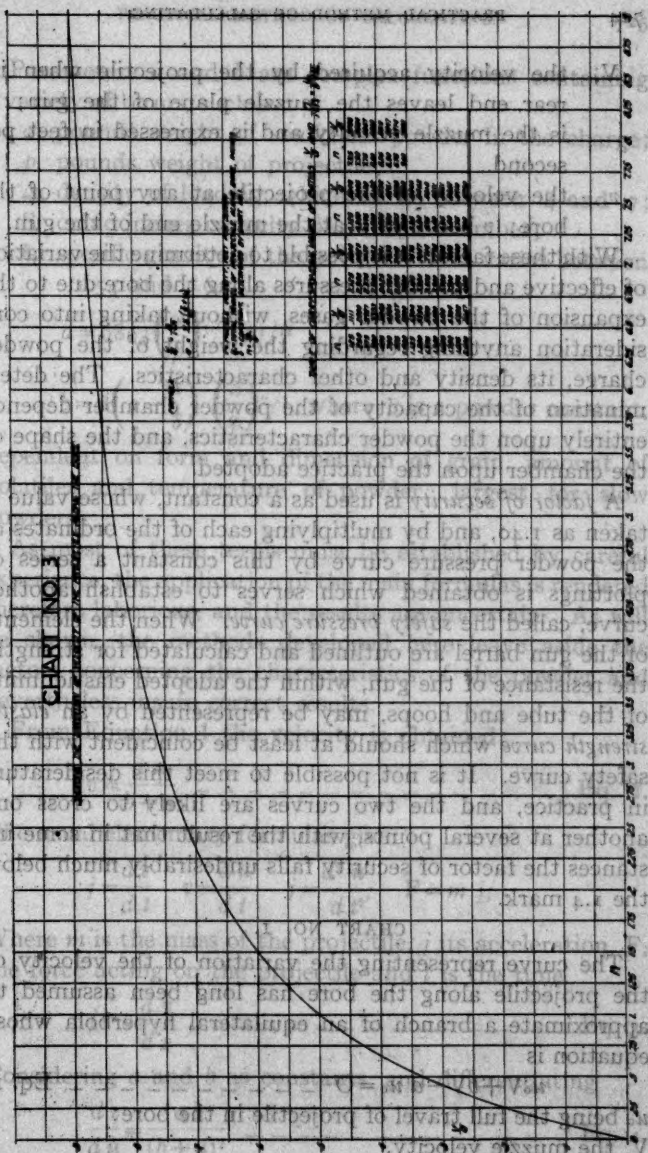
CHART NO. 3

V, the velocity acquired by the projectile when its rear end leaves the muzzle plane of the gun; it is the muzzle velocity and is expressed in feet per second.

v, the velocity of the projectile at any point of the bore; v becomes V at the muzzle end of the gun.

With these factors it is possible to determine the variation of effective and powder pressures along the bore due to the expansion of the powder gases, without taking into consideration anything regarding the weight of the powder charge, its density and other characteristics. The determination of the capacity of the powder chamber depends entirely upon the powder characteristics, and the shape of the chamber upon the practice adopted.

A *factor of security* is used as a constant, whose value is taken as 1.40, and by multiplying each of the ordinates of the powder pressure curve by this constant a series of plottings is obtained which serves to establish another curve, called the *safety pressure curve.* When the elements of the gun barrel are outlined and calculated for strength, the resistance of the gun, within the adopted elastic limits of the tube and hoops, may be represented by an *elastic strength curve* which should at least be coincident with the safety curve. It is not possible to meet this desideratum in practice, and the two curves are likely to cross one another at several points, with the result that in some instances the factor of security falls undesirably much below the 1.4 mark.

CHART NO. 1.

The curve representing the variation of the velocity of the projectile along the bore has long been assumed to approximate a branch of an equilateral hyperbola whose equation is

$$u_o V + bV - a u_o = 0 \qquad \text{Eq. 1,}$$

u_o being the full travel of projectile in the bore:
V, the muzzle velocity.

Parameters a and b are complex functions containing a variety of terms, such as:

ω, pounds weight of smokeless powder in the charge;

p, pounds weight of projectile;

Δ, density of loading, which varies between .4 and .7;

S, cubic inches capacity of powder chamber;

δ, specific gravity of the powder. This varies between 1.54 and 1.62.

$$a = 6823\left(\frac{\omega}{p}\right)^{1/2}\Delta^{1/12}$$

$b = \beta\left(1-\frac{\Delta}{\delta}\right)\left(\frac{S}{p}\right)^{2/3}$, where β is a powder constant, dependent on form and dimension of grain, amount of volatiles and temperature of powder; largest for slow powders.

As most of these terms must be established by careful experience, the application of the main formulas is rendered therefore laborious, and the results are uncertain. As will be shown, the methods developed here leave aside the factors concerning the characteristics of the powder and nevertheless insure correct results.

From Equation 1 the velocity is obtained:

$$v = \frac{a\,u}{b+u} \qquad \text{Eq. 2.}$$

The usual elementary formulas give:

$$j = \frac{d\,v}{d\,t} \quad v = \frac{d\,u}{d\,t} \quad j = \frac{d^2\,u}{d\,t^2}; \quad F = m\,j;$$

Where m is the mass of the projectile, j its acceleration, F, the force acting on the projectile and t is the time.

$$j = v\frac{d\,v}{d\,s} \qquad \text{Eq. 3.}$$

Considering a and b as constants, and differentiating

$$\frac{d\,v}{d\,u} = \frac{a\,b}{(b+u)^2} \qquad \text{Eq. 4.}$$

Replacing in Equation 3, v by its value in Equation 2 and $\frac{dv}{du}$ by its value in Equation 4, the acceleration is:

$$j = v\frac{dv}{du} = \frac{au}{(b+u)} \cdot \frac{ab}{(b+u)^2} = \frac{a^2bu}{(b+u)^3}.$$

Whence $F = mj = \frac{m a^2 b u}{(b+u)^3}$ - - - - - - - - - - - - Eq. 5.

Now, leaving a and b constant it is easy to show that F becomes a maximum when the travel $u = \frac{b}{2}$.

Replacing u by $\frac{b}{2}$ in Equation 5, and designating F, the maximum pressure, by F_m, we have

$$F_m = \frac{4}{27} m \frac{a^2}{b} \text{ - - - - - - - - - - - Eq. 6.}$$

Let F_e be a constant pressure acting upon the projectile throughout the travel u_o, (expressed in feet) at the end of which the projectile has acquired the velocity V, then

$$F_e u_o = \frac{mV^2}{2}.$$

Whence $F_e = \frac{mV^2}{2u_o}$ - - - - - - - - - - - - - - - - Eq. 7.

Squaring both members of Equation 2, making $v = V$ and $u = u_o$:

$$V^2 = \frac{a^2 u_o^2}{(b+u_o)^2}.$$

This value introduced in Equation 7 gives:

$$F_e = \frac{m}{2u_o} \cdot \frac{a^2u_o^2}{(b+u_o)^2} = \frac{m}{2} \frac{a^2u_o}{(b+u_o)^2} \text{ - - - - Eq. 8.}$$

Dividing F_e by F_m, from Equations 6 and 8:

$$\frac{F_e}{F_m} = \frac{m}{2} \cdot \frac{a^2 u_o}{(b+u_o)^2} \cdot \frac{27}{4} \cdot \frac{b}{ma^2} = \frac{27}{8} \cdot \frac{bu_o}{(b+u_o)^2} \quad \text{Eq. 9.}$$

Usually the maximum powder pressure P_b is a given quantity. The maximum pressure P_p, upon the projectile, is less than P_b, the difference being allowed to account for friction in the bore, in the rifling grooves, etc. The value adopted is $P_p = 1.12\ P_b$.

As F_m is the total pressure upon the projectile, A being the cross section of the bore

$$P_p = \frac{F_m}{A} = \frac{P_b}{1.12}$$

$$\text{Whence } F_m = \frac{AP_b}{1.12}$$

The cross section A is understood to be the total area of the bore and of the rifling grooves. In the preliminary project the number and shape of the grooves are not yet determined, therefore it is necessary to estimate approximately the cross sectional area of the grooves proper. This may be done by comparison with guns already made. An easy way is to take the average ratio of the total cross sectional area of the bore of the 51 guns, listed in the cannon table No. 1549, to the cross sectional area of the bore proper. Calling A_o the area of the bore, of diameter C, it is found that

$$\text{Mean ratio } \frac{A}{A_o} = 1.0242 \quad \text{Eq. 10.}$$

For calculation this may be rounded up to 1.025. This figure is adopted here.

Then, since $A_o = \frac{\pi C^2}{4}$, we have $A = \frac{1.025}{4}\pi C^2$, and

$$F_m = \frac{1.025}{4}\pi C^2 \cdot \frac{P_b}{1.12} = 0.71878\ C^2 P_b \quad \text{Eq. 11.}$$

Replacing F_e and F_m in Equation 9 by the values in Equation 7 and Equation 11, making $m = \frac{p}{g = 32.16}$, and taking u_o to be expressed in inches, we have

$$\frac{F_e}{F_m} = \frac{6pV^2}{32.16u_o} \cdot \frac{1}{0.71878C^2P_b} = \frac{27}{8} \cdot \frac{bu_o}{(b+u_o)^2}$$

Whence $$\frac{6\times 8}{32.16\times 0.71878\times 27} \cdot \frac{pV^2}{C^2P_b u_o} = \frac{bu_o}{(b+u_o)^2}$$

Or $$0.07691 \frac{pV^2}{C^2P_b u_o} = \frac{bu_o}{(b+u_o)^2} \quad \text{Eq. 12.}$$

Making the first member = constant = k, for the present, the equation may be put in the form:

$$k(b+u_o)^2 - bu_o = 0$$

$$kb^2 + 2kbu_o + ku_o^2 - bu_o = 0$$

$$b^2 + u_o \frac{(2k-1)}{k} b = -u_o^2$$

Solving for b:

$$b = -\frac{u_o(2k-1)}{2k} \pm \sqrt{u_o^2 \left(\frac{2k-1}{2k}\right)^2 - u_o^2} =$$

$$= u_o \left[\frac{1}{2k} - 1 - \sqrt{\left(\frac{1}{2k} - 1\right)^2 - 1}\right] \quad \text{Eq. 13.}$$

And finally

$$\frac{b}{u_o} = \left(\frac{1}{2k} - 1\right) - \sqrt{\left(\frac{1}{2k} - 1\right)^2 - 1} \quad \text{Eq. 14.}$$

The factors composing k being given quantities it is but a matter of simple calculation to obtain b without taking into account the powder characteristics.

By means of Equation 14 a diagram may be established which will give at once the ratio $\frac{b}{u_o}$ corresponding to a value of $2k$. All the preliminary work necessary will consist only in calculating $2k$.

Replacing k by its value in Equation 12, gives:

$$\frac{b}{u_o} = \left(6.5\,\frac{C^2 P_b u_o}{pV^2} - 1\right) - \sqrt{\left(6.5\,\frac{C^2 P_b u_o}{pV^2} - 1\right)^2 - 1} \quad \text{Eq. 15.}$$

In order to determine the coordinates of the diagram, let

$$6.5\,\frac{C^2 P_b u_o}{pV^2} = k_o \quad \text{Eq. 16.}$$

then $$\frac{b}{u_o} = (k_o - 1) - \sqrt{(k_o - 1)^2 - 1}$$

and $$\left[\frac{b}{u_o} - (k_o - 1)\right]^2 = (k_o - 1)^2 - 1$$

Whence $$k_o = \frac{\left(\frac{b}{u_o} + 1\right)^2}{2\,\frac{b}{u_o}} \quad \text{Eq. 17.}$$

Practice shows that $\frac{b}{u_o}$ may vary between the limits 0.150 to 0.650. By calculating the values of k_o corresponding to $\frac{b}{u_o} = 0.150,\ 0.175,\ 0.200,\ 0.225,\ \ldots\ 0.575,\ 0.600,\ 0.625,\ 0.650$, or, in all, 21 values, a curve may be established which will be found very convenient, and will suit any value encountered in practice. Such a curve is submitted herewith on Chart No. 1. This Chart is complete in itself, containing directions for use and an example of the computation neces-

sary as applied to one gun in service. The gun selected at random is the 4.7″ A.A. Model 1918. The given factors are:

$C = 4.7''$
$u_o = 166.13''$
$P_b = 34{,}000$ lbs. per sq. in.
$p = 45$ lbs.
$V = 2400$ ft. per second.

$$k_o = \frac{6.5 \times 4.7^2 \times 34000 \times 166.13}{45 \times 2400^2} = 3.129$$

This value is easily read on the k_o scale of the Chart and when plotted horizontally on the curve the vertical projection of this plotting onto the lower scale gives at a glance the required ratio $\frac{b}{u_o}$ which, in this instance is 0.2495.

Then $b = 0.25$ and $u_o = 0.2495 \times 166.13''$ and the point of maximum pressure, as derived from Equation 5, corresponding to $\frac{b}{2}$ as abscissa, is situated at $\frac{b}{2} = \frac{0.2495 \times 166.13}{2} =$ 20.745″ from the origin of the travel of the projectile in the bore.

VARIATION OF PRESSURE ALONG THE BORE—CHART NO. 2.

Reverting to Equation 5, and making $F_x = F$ = pressure, in pounds per square inch, for any length of travel u, then

$$F_x = ma^2b \frac{u}{(b+u)^3}$$

The maximum value of F_x is F_m; it corresponds to a travel $\frac{b}{2}$ and is a known quantity. It is expressed in Equation 6 as

$$F_m = \frac{4}{27} m \frac{a^2}{b}$$

Dividing F_x by F_m will eliminate m and a^2 and will make it possible to calculate P_x for any value of u by means of known constants.

Thus $$\frac{F_x}{F_m} = ma^2b\frac{u}{(b+u)^3} \cdot \frac{27}{4}\frac{b}{a^2m} = \frac{27}{4}b^2\frac{u}{(b+u)^3} \quad \text{Eq. 18.}$$

Both F_x and F_m represent total pressures acting upon the full rifled bore area. Dividing each by this area gives the corresponding pressure per square inch, respectively P_x and P_m.

Therefore

$$\frac{P_x}{P_m} = \frac{F_x}{A} \cdot \frac{A}{F_m} = \frac{27}{4}b^2\frac{u}{(b+u)^3} \quad \text{Eq. 18}^1.$$

Let $\frac{u}{b} = n$, then $\frac{u}{(b+u)^3} = \frac{nb}{(b+nb)^3} = \frac{n}{b^2(1+n)^3}$ and

$$\frac{P_x}{P_m} = \frac{27}{4}\cdot\frac{n}{(1+n)^3} \quad \text{Eq. 19.}$$

By giving to n successively increasing values, up to a practical limit, and solving for each, a series of $\frac{P_x}{P_m}$ ratios may be had which can be conveniently charted for use. In this way Chart No. 2 was established, and it makes it possible to read at once the ratio $\frac{P_x}{P_m}$ corresponding to a given ratio of $\frac{u}{b}$.

To compute this chart more easily the difference between two consecutive values of n was selected as $\frac{1}{8}$ for the convenience of the delineator between 0 and 5, and as $\frac{1}{4}$ between 5 and 9, the limit of the Chart.

Whence, making

$$n = \frac{1}{8}, \frac{2}{8}, \frac{3}{8}, \frac{4}{8}, \frac{5}{8},$$

we have $\frac{P_x}{P_m} = \frac{27}{4}\frac{n}{(1+n)^3} = 432\frac{1}{(9)^3}, 432\frac{2}{(10)^3}, 432\frac{3}{(11)^3},$

$$432\frac{4}{(12)^3}, 432\frac{5}{(13)^3}.$$

The values thus computed are tabulated below. In many instances they will be found sufficient for the purpose, if the length of travel is suitably divided, and will save a little time which otherwise would be required for reading them on the Chart.

n	$\frac{P_x}{P_m}$	n	$\frac{P_x}{P_m}$	n	$\frac{P_x}{P_m}$	n	$\frac{P_x}{P_m}$
.125	.593	2.125	.471	4.125	.207	7.25	.0872
.250	.864	2.250	.442	4.250	.1982	7.50	.0824
.375	.974	2.375	.417	4.375	.1905	7.75	.078
.500	1.000	2.500	.394	4.500	.1826	8.00	.074
.625	.983	2.625	.372	4.625	.1758	8.25	.0702
.750	.945	2.750	.352	4.750	.1686	8.50	.0669
.875	.896	2.875	.334	4.875	.1623	8.75	.064
1.000	.844	3.000	.315	5.000	.1562	9.00	.0607
1.125	.792	3.125	.300	5.25	.1455		
1.250	.741	3.250	.286	5.50	.1352		
1.375	.693	3.375	.272	5.75	.1271		
1.500	.648	3.500	.259	6.00	.118		
1.625	.606	3.625	.248	6.25	.1112		
1.750	.568	3.750	.236	6.50	.104		
1.875	.533	3.875	.226	6.75	.098		
2.000	.500	4.000	.216	7.00	.0923		

When a suitable number of $\frac{P_x}{P_m}$ ratios have been selected for a given gun the corresponding powder pressure is obtained for each ratio by simply multiplying the latter by the P_m given, or $P_x = \text{ratio} \times P_m$. The pressures thus obtained may be plotted at a convenient scale and a curve may be drawn, joining all the plottings; this is the *powder pressure curve.*

By multiplying each of the ordinates of this curve by the constant 1.4, new plottings may be obtained which when joined will give the Safety Pressure Curve. The computation of the ordinates of this and the preceding curve may be made on the slide rule with an accuracy amply sufficient for ordinary practice.

Chart No. 3—Showing the variation of velocity of the projectile along the bore.

Taking Equation 2, $v = a\frac{u}{b+u}$, and solving for a when $v = V$, $u = u_o$ and with b obtained by means of Chart No. 1,

we have $a = \frac{V \times (b+u_o)}{u_o}$ which makes it possible to compute the velocity V_x for any travel u_x.

Making $n = \frac{u}{b}$, just as was done for Chart No. 2.

$$v = a\frac{n}{1+n} \quad \text{Eq. 20.}$$

By giving to n successive values from $\frac{1}{8}$ to 9 a series of numbers is obtained which, when each is multiplied by the constant a for a given gun, will give all the ordinates necessary to determine the plotting of the *velocity curve*. Chart No. 3 shows a curve giving the values of $\frac{v}{a}$ corresponding to $\frac{n}{1+n}$ between the limits 0 to 9.

The computed values of $\frac{n}{1+n}$ are given in the following table. In many instances they may be used directly, thus saving reference to the Chart.

n	$\frac{v}{a}$	n	$\frac{v}{a}$	n	$\frac{v}{a}$	n	$\frac{v}{a}$
.125	.1111	2.125	.6800	4.125	.8049	7.25	.8788
.250	.2000	2.250	.6923	4.250	.8095	7.50	.8824
.375	.2727	2.375	.7037	4.375	.8139	7.75	.8857
.500	.3333	2.500	.7143	4.500	.8182	8.00	.8889
.625	.3846	2.625	.7241	4.625	.8222	8.25	.8919
.750	.4286	2.750	.7333	4.750	.8261	8.50	.8947
.875	.4667	2.875	.7419	4.875	.8298	8.75	.8974
1.000	.5000	3.000	.7500	5.000	.8333	9.00	.9000
1.125	.5294	3.125	.7576	5.25	.8400		
1.250	.5556	3.250	.7647	5.50	.8462		
1.375	.5789	3.375	.7714	5.75	.8518		
1.500	.6000	3.500	.7778	6.00	.8571		
1.625	.6190	3.625	.7838	6.25	.8621		
1.750	.6364	3.750	.7895	6.50	.8667		
1.875	.6522	3.875	.7949	6.75	.8710		
2.000	.6667	4.000	.8000	7.00	.8750		

NOTES.

THE NEW CAPITAL SHIPS.

Mr. Lloyd George has frankly told Parliament that the replacement of four obsolete capital ships of the British Navy is being undertaken quite irrespective of the possible success or failure of the forthcoming Washington Conference. There has been a certain amount of Parliamentary pressure on the Government to delay the giving out of the tenders for the four new capital ships beyond September, but great as is the financial stringency, the decision of the Government is against further delay. The laying down of these four ships is indeed no stimulation of competitive building, if we have regard to what America and Japan have been doing in the way of expanding their fleets. Our whole attitude since the armistice has been to hold back rigorously. At the same time, as the Prime Minister says, it would be a dereliction of duty to allow the efficiency of the Navy to deteriorate or to neglect to provide it with material in which it can feel confidence. Our security at sea must be maintained, and it is better to proceed with the preparations for four new ships quietly and at moderate speed than risk an outburst of popular agitation later that we were neglecting our defences. It is just as well to enlist public confidence by timely, which need not be excessive, preparation.—"Shipbuilding and Shipping Record," July 28, 1921.

NAVAL STRENGTH OF THE POWERS.

The secondary position which this country now occupies among the naval powers of the world, is clearly shown by a Government return recently made to the House of Commons by the Secretary of the Admiralty. The statistics, which are summarized in the accompanying tables, include warships of the classes indicated, which have been launched since the end of 1900 and also refer to vessels under construction. In the matter of capital ships it will be seen that the first place is now held by the United States, which country has 36 battleships built and 11 battleships and 6 battle cruisers under construction, while Great Britain has 23 battleships and 8 battle cruisers, but is not actually building any capital ships at the present time. Japan, which now takes third place as a naval power, is building 3 battleships and 2 battle cruisers and even Russia is believed to have 4 ships of the latter class in hand.

With regard to the British capital ships, it should be mentioned that the latest battleships are those of the *Royal-Sovereign* class, including *Royal Sovereign, Royal Oak, Revenge, Resolution* and *Ramillies*, all of which were completed in 1916, except the *Ramillies*, which was finished in the following year. None of these vessels has therefore been designed since the lessons of the war have been assimilated. British battle cruisers, excepting the *Hood, Renown* and *Repulse*, were all completed before the war, and, of the three exceptions, only the first-mentioned is sufficiently modern to include design features arising from war experience. The United States, on the other hand, has completed the battleships *Mississippi, New*

Mexico, Idaho and *Tennessee* since 1917, and the 11 now building will be completed before the end of next year. Six of the latter, viz., those of the *South Dakota* class, are of 43,200 tons displacement and of 60,000 horsepower, and each carries twelve 16-inch guns, sixteen 6-inch guns, four 14-pounder anti-aircraft guns, and two 21-inch submerged torpedo tubes. The battle cruisers now building in the United States are of the *Lexington* class and are expected to be completed in 1924. They are of about the same displacement as the battleships, viz., 43,500 tons, but the machinery, which is of the turbo-electric type, will be of 180,000 horsepower. The armament is similar to that of the battleships, except that only eight 16-inch guns are carried and that there are four additional torpedo tubes above water as well as two extra submerged tubes.

Of the battle cruisers building in Japan, no particulars are at present available, and all that is known of the Japanese battleships under construction is that one of them, the *Mutsu*, is of 33,800 tons displacement and 46,000 horsepower and that she will carry eight 16-inch guns, twenty 5.5-inch guns, four 12-pounder high-angle guns, and eight torpedo tubes. The completed warships include *Fuso, Yamashiro, Ise, Hyuga* and *Nagato*, all finished since the outbreak of war. The latest of these, the *Nagato*, which was only completed last year, carries the same armament as the *Mutsu*, and is also of the same horsepower and displacement.

Although Russia is nominally building four battle cruisers, two of which were launched in 1915 and the other two in 1917, the work on these vessels is not actually in progress. Their tonnage is given as 32,500 and their horsepower as 68,000, while the armament is stated to be twelve 14-inch guns, twenty-four 5.1-inch guns, and six submerged torpedo tubes. Whether these vessels will ever be completed is, perhaps, open to question considering the internal condition of the country, but, in any case, the vessels are outclassed by those of other powers above referred to. The point we wish to emphasize in connection with capital ships is that the position of Great Britain is relatively weaker than appears to be the case from a consideration of numbers only, owing to the fact that the vessels completed, or approaching completion, in the United States and Japan are of later design.

No country is now building the class of vessels known as armored cruisers and first-class protected cruisers, but the British Navy includes three of this type—*Courageous, Glorious* and *Antrim*—while the United States possesses 15 and Japan 4. France, Russia and Italy have 10, 9 and 5 cruisers, respectively, but 2 of the Italian cruisers are regarded as of no military value. The position with regard to light cruisers is given in Table I, from which it will be seen that, of this class of vessel, Great Britain owns more than all the rest of the world put together. The British light cruisers building are three of an improved *Birmingham* class, three "D" class, two "E" class, and two other vessels. With regard to aircraft carriers the British Navy also maintains its foremost position, having four vessels of this type completed and two in hand. The completed aircraft carriers are *Furious, Argus, Pegasus* and *Ark Royal*, while those building are *Hermes* and *Eagle*. The United States and France possess one aircraft carrier each, and both these countries have another vessel of the same class in hand. Japan has no aircraft carriers at present, but the construction of one is under consideration.

With regard to smaller craft, the relative positions of the various naval powers can be seen from Table II, and the predominant position of the United States, both in the matter of destroyers and submarines, will at once be evident. The United States Navy includes 278 destroyers, of which 178 have been completed since the end of 1918. Of the 40 now under construc-

TABLE I.—CAPITAL SHIPS AND LIGHT CRUISERS.

	Battleships.		Battle-Cruisers.		Light Cruisers.	
	Built.	Building.	Built.	Building.	Built.	Building.
Great Britain	23	...	8	...	51	10
United States	36	11	...	6	15	10
Japan	12	3	7	2	9	8
Italy	12*	...	...	...	10	...
France	11	...	...	...	5	6
Russia	13	1	...	4	1	(Projected). 8
Germany	8	...	...	...	8	...

*The Minister of Marine has power to remove 4 of these vessels from the list.

TABLE II.—SMALL CRAFT.

	Flotilla Leaders.		Destroyers.		Submarines.	
	Built.	Building.	Built.	Building.	Built.	Building.
Great Britain	16	2	185	6	89	8
United States	...	...	278	40	103	46
Japan	...	...	84	9	23	15
Italy	8	3	52	12	64	3
France	1	1	70	1 (12 projected),	48	6
Russia	....	....	98	21	36	22
Germany	...	...	16	12	...	...

tion, nearly all are of 1,315 tons displacement, and of 27,000 horsepower, while the armament is generally four 4-inch guns, one 14-pounder anti-aircraft gun, and four 21-inch triple torpedo tubes. Only six destroyers and two flotilla leaders are now building in Great Britain, the two flotilla leaders being *Keppel* and *Rooke*, both of 1,750 tons displacement and of 40,000 horsepower. The armament of these vessels includes five 4.7-inch guns, one 3-inch gun, two 2-pounder guns and two triple torpedo tubes. The British destroyers under construction are *Shikari*, *Thracian*, *Whitehall*, *Worcester*, *Wren*, and *Witch*. The two first-mentioned are of 1,075 tons displacement and of 27,000 horsepower, and both carry three 4-inch guns, one 2-pounder gun and two double torpedo tubes; the other four displace 1,325 tons and carry four 4.7-inch guns, two 2-pounder guns, and two triple torpedo tubes. The horsepowers of *Whitehall*, *Worcester* and *Wren* are 27,000, while that of *Witch* is 30,000.

Russia, it will be seen, takes third place with regard to destroyers, having 98 of these vessels completed and 21 in hand, while the Japanese Navy, which occupies the fourth place, includes 84 destroyers and is building

nine in addition. The latter include six first-class and four second-class vessels, the former displacing 1,345 tons and having machinery of 38,500 horsepower. The armament of the first-class destroyers is four 4.7-inch guns, two machine guns and six torpedo tubes; six exactly similar vessels have already been completed.

In connection with submarines, the extensive naval development of the United States will also be apparent from Table II. There are 46 submarines now building for the United States Navy, which already includes 103, and the new vessels will all be completed during this year or next. The majority of the existing vessels are also of late design, only 33 of them having been completed before 1918. The British Navy, as shown in the Table, includes 89 submarines and is building eight in addition. Of the vessels building, seven are of the "L" type and one is of the "K" type. In the matter of submarines it will be seen that the Japanese Navy is relatively weak, since it includes only 23 vessels of that type, while 15 additional submarines are building. Italy, on the other hand, has 64 submarines and is adding three to their number, while Russia is building 22, so that, if these were to be completed in a reasonable time, the Russian submarine Navy would be greater than that of either France or Japan. Very little information is, however, available as to the Russian Navy, and the figures given, both for vessels building and completed, are by no means reliable.

The condition of impotence to which the German Navy has been reduced can be gathered from the tables, and to this we may add the information that the eight battleships it now includes, are all old designs completed between 1904 and 1908; no guns larger than 11-inch are mounted in them. The German light cruisers are even older, dating from 1899 to 1905, and their armament includes ten 4.1-inch guns in each case, except one, which has been completely disarmed. The most modern destroyers remaining in the German Navy were completed in 1913 and are of 555 tons displacement and of 15,000 horsepower; they carry two 4.4-inch guns, four machine guns and four torpedo tubes. The only other war vessels left to Germany are 16 torpedo boats, completed between 1907 and 1911 and at present disarmed.

Thus, what was in pre-war times the main incentive to naval construction in this country has, for the time being at all events, ceased to exist. Sea power is, however, still of vital importance to the integrity of a scattered empire such as ours, and the British Navy is today, no less than formerly, an essential instrument for the preservation of the peace of the world. Thus, although the necessity for numerical superiority may be less than it was, the need for the highest possible efficiency is even greater. For this reason, it is important for the public to realize the fact, above referred to, that, whereas the majority of the capital ships of the British Navy were designed before lessons learned in the war could be applied, some foreign powers have made, and are making, considerable use of the recent practical experience in naval warfare.—"Engineering," July 29, 1921.

AERO ENGINES.

By Alan E. L. Chorlton, C.B.E., M.Inst.C.E., M.I.Mech.E.

Lecture I.—Delivered January 17th, 1921.

Mechanical flight has been a problem of great attraction from very early times. Though most early writers and experimenters conceived the prime mover to be man himself, reasoning from an analogy to bird flight, the few even long ago arrived at the conclusion that some more powerful

motor than human muscles was practically a necessity. Giovanni Borelli, in 1680, says: "It is impossible that men should be able to fly craftily by their own strength," and Sir George Cayley, 1809, whilst admitting that man "if he can be made to exert his whole strength advantageously upon a light surface similarly proportioned to his weight as that of the wing of a bird * * * would fly like a bird." However, he says, "To produce this effect (practical flying with considerable weight) it is only necessary to have a first mover which will generate more power in a given time, in proportion to its weight, than the animal system of muscles."

Though Sir George Cayley, judging by his qualification only, apparently did not think the prime mover to be a great difficulty it proved to be so in effect, for we successfully flew on gliders before we had developed the first mover to impel them.

In these early days Sir George Cayley saw that the only prime mover then available, the steam engine, was not the only agent, for he says, after dealing with possible developments of Watt's steam engine, "It may seem superfluous to inquire further relative to a first mover for aerial navigation, but lightness is of so much value in this instance that it is proper to notice the probability that exists of using the expansion of air by the sudden combustion of inflammable powders or flints with great advantage," instancing an early oil engine of William Chapman, of Newcastle.

The earliest internal combustion engine described in this country, however, appears to have been that of Robert Street, in 1794. This motor contained a cylinder heated at the bottom by a fire to evaporate spirits of turpentine below a motor piston. The motor piston was drawn up and air allowed to enter to mix with the vapor, which was then ignited driving the piston further upwards and forcing a pump piston up by means of a lever, the return stroke being performed by the weight of the motor piston. Sir George Cayley was evidently aware of the advantages which might be obtained with this type of engine, although he apparently did not know of the proposal of Street, for he states, "Probably a much cheaper engine might be produced by a gas light apparatus and by firing the inflammable air generated with a due proportion of common air under a piston." The last sentence represents more or less exactly the aero engine we have today, carburetted air mixed with ordinary air fired electrically against a piston. However, the successful aero engine was to be long in coming, so long that the steam engine was still the only possible agent for experimenters who had arrived at the stage of completing their designs, up to the prime mover, until practically the present day.

In searching for possible means for producing motive power other than the steam engine, the early investigators considered a very great number of possible combustible materials for utilization by the internal combustion method; both gaseous, liquid and solid materials were contemplated, but up to the present, success has only been achieved by the use of gaseous and liquid fuels. The relative value of a few of the different substances that have been proposed for the purpose of producing power depends, of course, primarily on the quantity of heat evolved on combustion. The calorific values for a number of different materials are given in Table I, the products of combustion being assumed to be carbon dioxide, sulphur dioxide and water in the state of vapor. The heat values are given in British thermal units per pound of the material in each case.

TABLE I.—CALORIFIC VALUE OF DIFFERENT SUBSTANCES. BRITISH THERMAL UNITS PER POUND.

Hydrogen	34,800
Marsh Gas	23,500
Acetylene	21,450
Amyl Alcohol	16,100
Ethyl Alcohol	12,900
Methyl Alcohol	9,600
Benzol	17,900
Crude Petroleum	19,800
Petrol	19,800–20,100
Bituminous Coal	11,000–14,000
Anthracite	12,600–13,400
Coke	12,600
Pine containing 12 per cent of water	7,950
Oak containing 12 per cent of water	7,150
75 per cent Dynamite	2,320
Gunpowder	1,300
Explosives of Ammonium Nitrate Class containing 83 per cent of Ammonium Nitrate	1,470
Trinitro Toluol. (T.N.T.)	6,498

From the above Table it will be seen that the fuels which are in practice used in internal combustion engines are some way down the scale as regards possible heat value per unit weight of fuel. The advantage on this basis of utilizing a fuel with high hydrogen content will be evident from the Table. The practical difficulties to be expected are great; but as light weight is of such fundamental importance in flying the question of the production of a higher value fuel seems to merit consideration.

Explosive compounds such as are used in guns could be applied in cartridge form to the turbo engine, but the practical difficulty, the weight, since the oxygen for combustion is to a great extent to be carried with the compounds, renders their use prohibitive.

The use of steam in turbines, instead of internal combustion engines, is advocated on the score of greater reliability, but it must not be forgotten that this reliability with high efficiencies has only been obtained on land plants, where weight considerations could to a great extent be disregarded. The great disadvantage of generating heat outside the working fluid is, of course, that it involves the use of a boiler in addition to the motor itself, and, if high efficiencies are to be obtained, a superheater and an efficient condenser. The latter alone, on account of the necessity for expanding to very large volumes and for using air-cooling for condensation with the consequent large bulk and head resistance, would prevent any approach to the highest possible turbine efficiencies for flight purposes. On this account even the most recent developments of the steam turbine cannot compete with the modern aero engine as regards weight for power. As regards thermal efficiency, the steam turbine plant is still considerably behind the modern aero internal combustion engine plant. For low engine weight, moreover, it would be necessary to employ high speed impulse type turbines, and this again would involve the use of gearing; it is probable that a low weight steam turbine plant would only be obtained at the expense of considerable reduction in thermal efficiency. Hence it appears that the use of the steam turbine for aerial work is bound to be a limited one.

The first successful steam turbine was running about 1884, driving a dynamo and developing about 10 horsepower at a speed of 18,000 revolutions per minute.

A great step forward was made about 1890, and the first condensing steam turbine was produced in 1892, developing 200 horsepower, at a speed of 4,800 revolutions per minute, with a steam consumption of 16 pounds per indicated horsepower hour. This marked a great advance in steam turbine construction, and since that date there has been continuous improvement in efficiency, until at the present time it is considered by Sir Charles Parsons that overall brake thermal efficiencies approaching 28 per cent are within reach of attainment in large plants of the order of 25,000 brake horsepower.

TABLE 2.

THERMAL EFFICIENCY TO HEAT IN FUEL.

Type of Engine.	Date.	Particulars of Engine.	Thermal Efficiency.	
			Indicated.	Brake.
Steam Engines	1884	Compound Reciprocating	10.8	9.7
	1892	First Condensing Steam Turbines	9.2	7.3
	1904	Condensing Steam Turbines	14.3	12.1
	1914	Condensing Steam Turbines	19.6	16.6
Gas Engines. Efficiency Calculated from Heat in Coal with assumed Efficiency of 85% for producer	1884	Crossley. 8.5" x 14"	14.5	12.2
	1892	Crossley. 8.5" x 18"	19.4	14.8
	1904	National. 14" x 22"	29.9	25.4
	1908	Crossley. 11.5" x 21"	31.4	27.2
Heavy Oil Engines	1890	Priestman. 8.5" x 12"	12.2	10.5
	1894	Crossley. 7" x 15"	17.5	15.6
	1903	Diesel. 15.75" x 23.6"	39.7	32.6
	1910	Mirrlees-Diesel 12" x 18.25"	43.4	31.6
	1920	Ruston High Compression	41.3	35.1

The use of the uniflow steam engine may possibly be considered as an alternative to the steam turbine.

The above table, number two, gives some average thermal efficiencies of internal combustion engines and steam turbines since 1884, and shows clearly the progress which has been made in this respect in both forms of producing motive power.

In the development of the steam turbine and the internal combustion engine for general purposes, the question of weight was not of primary importance; thus the weights were generally much in excess of those which we have become accustomed to in aerial work. In the aero engine the fullest advantage has been taken of the very extensive research which has been carried out with almost all possible materials of construction for the different parts of the engine.

The weight of the engines given in Table 2 varies from 150 pounds to 500 pounds per B.H.P., according to the different types.

Some indication of the weight of steam turbine plants as compared with internal combustion engines may be gathered from the fact that the total weight of turbines and shafting in the *Turbinia*, built in 1896, was 4.00 pounds per I.H.P. In this plant the turbines ran at the high speed of 2,300 revolutions per minute.

In the stationary motive power plants noted, the materials used were practically only cast iron, wrought iron and steel, all of which were of low tensile value. At the present date the tensile values of steels used for aero

engines are as high as 100 tons per square inch; the advantage of this is obvious. Aluminium alloys have also made enormous progress in recent years, having replaced iron and steel for many parts used in aero engine construction.

The specific gravities of various materials are given in Table 3.

TABLE 3.—SPECIFIC GRAVITIES OF VARIOUS METALS.

Lead	11.0
Copper	8.9
Nickel	8.7
Bronze	8.7
Steel	7.8
Wrought Iron	7.7
Cast Iron	7.5
Manganese	7.4
Tin	7.0
Zinc	7.0
Chromium	6.8
Antimony	6.6
Aluminium (Commercial)	2.6
Magnesium	1.75

In this table only specific gravities of elements are given; in practice various alloys are employed and the specific gravity of any particular alloy will be readily obtained when the different percentages are known.

In the early attempts at producing a motor of sufficiently high power-weight ratio to lift a heavier-than-air machine from the ground, thermal efficiency was naturally regarded as of secondary importance, and it is of interest to note a few of the steam engines which were produced by the early inventors.

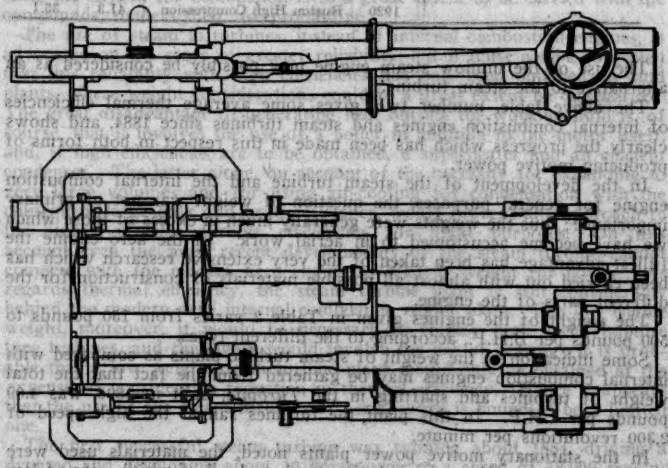

FIG. 1.

Langley produced a high pressure steam engine for a model aeroplane stated to give 1¼ horsepower with an engine weight of only 1.625 pounds, and a boiler weight of 5 pounds, so that the total weight of the power unit would be 6.625 pounds or 4.4 pounds per horsepower; but, as would be expected, no practical advance in power-propelled machines resulted from these experiments.

Maxim, in 1895, produced a steam plant employing a high pressure compound engine, Fig. 1, which had a high pressure cylinder 5 inches in diameter and a low pressure cylinder 8 inches in diameter, with a stroke of 12 inches. A tubular boiler was employed generating steam at a pressure of 320 pounds per square inch, the boiler being heated from a petrol furnace. The total weight of the boiler, furnace and engine was 2 000 pounds; the capacity of the plant was 350 horsepower, so that the weight amounted to 5.7 pounds per horsepower.

None of these early steam plants designed with a view to minimizing weight per horsepower attained any practical success and the attempts are only of historical interest.

Before discussing the thermodynamic cycles which are used in successful aero engines at the present day and those which give promise of successful development for special engines, it will be of interest to note the different ones which were used in the early development of the internal combustion engine.

In Type 1, used by Lenoir, the combustible charge was ignited at atmospheric pressure with the piston in the center and expanded adiabatically to atmospheric pressure or a pressure slightly above.

In a variation of this type the heat is added at constant volume and adiabatic expansion is continued to the lowest temperature, the heat being rejected at constant temperature so that during the instroke of the piston the line traced on the diagram is isothermal.

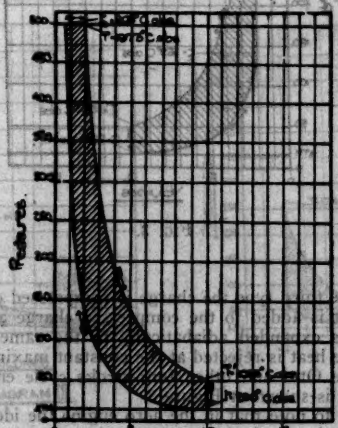

FIG. 2.

In the above type there is no compression of the charge before ignition; the efficiency is very low, but engines working on these cycles did attain considerable commercial success.

In all the later engines the charge is compressed before ignition from atmospheric pressure to some higher pressure and ignition takes place at the higher pressure.

In Type 2 the charge is compressed adiabatically from atmospheric pressure to the maximum pressure. It is then ignited and heat added at constant pressure. Adiabatic expansion then takes place until atmospheric pressure is reached and the heat is rejected during the instroke of the piston at atmospheric pressure.

In a modification of this type the expansion is stopped before atmospheric pressure is reached, the heat being rejected at constant volume. A diagram of this is shown at Figure 2. This type is the ideal cycle of the Brayton, Diesel and other constant pressure engines.

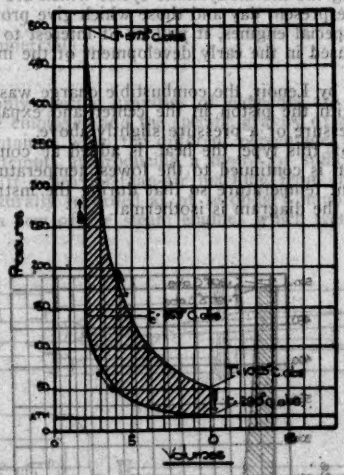

FIG. 3.

According to the third type the charge is compressed during the instroke of the piston, heat is added to the compressed charge at constant volume, and the charge is expanded adiabatically to the same volume as before compression. The heat is rejected at the constant maximum volume. This is the type of the Otto and other four-stroke cycle engines. The corresponding diagram is shown at Figure 3.

It is of interest to note that in the aero engine the ideal diagram at altitude is modified owing to the low atmospheric pressure and lower temperature. The effect of the lowering of the pressure at altitude and the lowering of the temperature are shown separately and combined at Figure 4.

In diagram I, the inlet pressure and temperature are those existing at ground level; in diagram II, ground level pressure is taken with the average temperature at 20,000 feet; in diagram III the pressure is the average pressure at 20,000 feet with ground level temperature and in diagram IV the average pressure and temperature at 20,000 feet are taken.

According to a modification of Type 3, the expansion may be carried further than the original volume, to atmospheric pressure or some higher pressure. This cycle, expanding to a volume greater than the original volume and rejecting heat partly at a constant volume from a pressure greater than atmospheric pressure and partly at constant atmospheric pres-

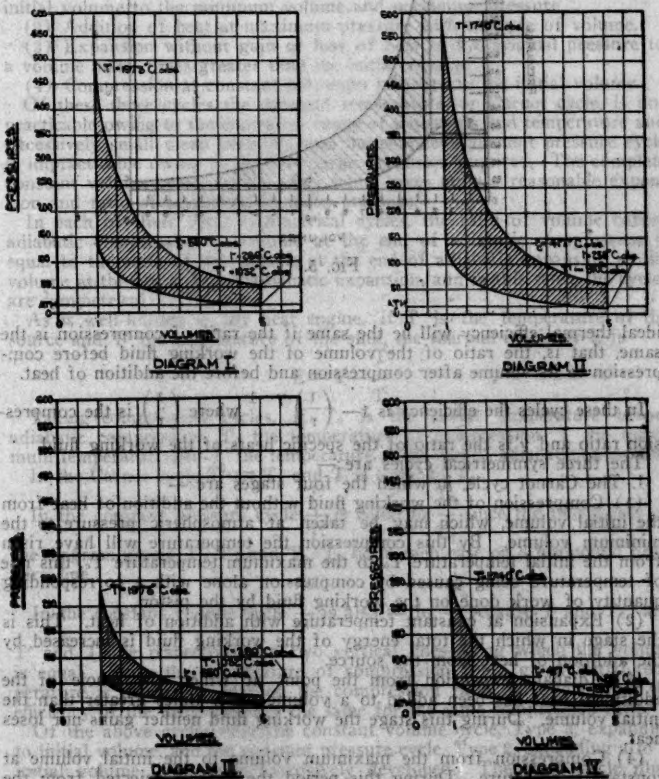

FIG. 4.

sure, is shown at Figure 5. The cycle has been employed, as will be shown later, and may be of use in the future development of aero engines.

It may be remarked that there are three symmetrical thermodynamic cycles according to which a heat engine, in which the working fluid is compressed before heat addition takes place, can operate. In all of these, the

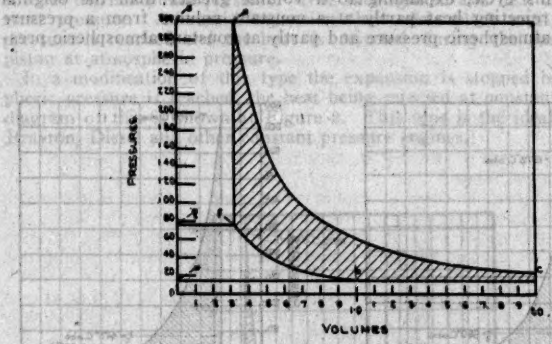

FIG. 5.

ideal thermal efficiency will be the same if the ratio of compression is the same, that is, the ratio of the volume of the working fluid before compression to its volume after compression and before the addition of heat.

In these cycles the efficiency is $1-\left(\frac{1}{r}\right)^{\gamma-1}$ where $\left(\frac{1}{r}\right)$ is the compression ratio and γ is the ratio of the specific heats of the working fluid.

The three symmetrical cycles are:—

I. The Carnot cycle, in which the four stages are:—

(1) Compression of the working fluid without the addition of heat from the initial volume, which may be taken at atmospheric pressure to the minimum volume. By this compression the temperature will have risen from the initial temperature T_o to the maximum temperature T_c, this rise of temperature being caused by compression alone with a corresponding quantity of work done on the working fluid by the piston.

(2) Expansion at constant temperature with addition of heat. This is the stage in which the total energy of the working fluid is increased by the addition of heat from the source.

(3) Adiabatic expansion from the point at which the whole of the additional heat has been added to a volume many times greater than the initial volume. During this stage the working fluid neither gains nor loses heat.

(4) Compression from the maximum volume to the initial volume at constant temperature. During this period the heat is rejected from the working fluid and absorbed by the sink. The constant efficiency is determined by the ratio of the volume before adiabatic compression on the first stage to the volume at the end of this adiabatic compression, and this volume ratio is equal to the ratio of the volumes of adiabatic expansion.

II. The constant volume cycle. This is the well-known cycle of the ordinary Otto four-cycle engine. The working fluid is first compressed without addition of heat, then heat is added at constant volume, the working fluid is expanded from the minimum volume to the initial volume without gain or loss of heat, and heat is rejected at constant volume.

III. The constant pressure cycle. This cycle is the perfect cycle on which it was at one time commonly stated that the Diesel and like engines worked. It will be seen from the diagram, however, that for practical purposes the high efficiency of this cycle is unattainable owing to the very considerable cylinder volume required for a given maximum pressure and heat addition. The four stages are:—

(1) Compression without heat addition from the initial pressure and initial volume to the minimum volume and maximum pressure.

(2) Addition of heat at maximum pressure with increase of volume.

(3) Expansion without gain or loss of heat to the original pressure to a volume many times greater than the initial volume.

(4) Compression at constant minimum pressure to the initial volume.

Of these three cycles the constant temperature, or Carnot cycle, is impracticable owing to the enormous range of expansion and temperature and excessively small mean pressure, and the complete constant pressure cycle is impracticable owing to the very large expansion required. The complete constant volume cycle, on the other hand, gives quite a reasonable expansion and ratio of maximum to mean pressure.

In each of these three symmetrical cycles, the ratio of volume before adiabatic compression to volume at the end of adiabatic compression is equal to the ratio of the volume at the end of adiabatic expansion to the volume at the beginning of adiabatic expansion, and in this sense the cycles are symmetrical.

As is well-known in any heat engine, if T is the temperature of the source and T′ the temperature of the sink, the thermal efficiency E is

$$E = 1 - \frac{T'}{T}$$

Taking compression cycles, if T_o is the initial temperature before adiabatic compression, T_c the temperature of compression, T the maximum temperature and T′ the temperature at the end of adiabatic expansion.

In the Carnot cycle $T' = T_o$ and T is equal to T_c.

In the constant pressure cycle $\frac{T'}{T}$ is equal to $\frac{T_o}{T_c}$ since the two points on the diagram giving T′, T; and T_o, T_c lie in each case on a horizontal pressure line.

In the constant volume cycle $\frac{T'}{T}$ is also = to $\frac{T_o}{T_c}$ because T and T_c and T′ and T_o lie respectively on two vertical lines along which the volume is equal. Since the temperature of compression for a given initial temperature is dependent only on the compression ratio, it follows that the efficiency of the three cycles is the same for the same compression ratio.

Of the above cycles only the constant volume cycle, Type 3, expanding to initial volume, and the constant pressure cycle, Type 2, expanding also to initial volume, have been extensively developed. The third cycle illustrated by Type 2, expanding beyond initial volume, was used in the Atkinson cycle engine in 1886. An ingenious link motion was employed to obtain a variable stroke of the piston. The link motion is illustrated at Figure 6. An engine of this type was tested at the Trials organized by the Royal Society of Arts in 1888, and developed a brake horsepower of about 9.5

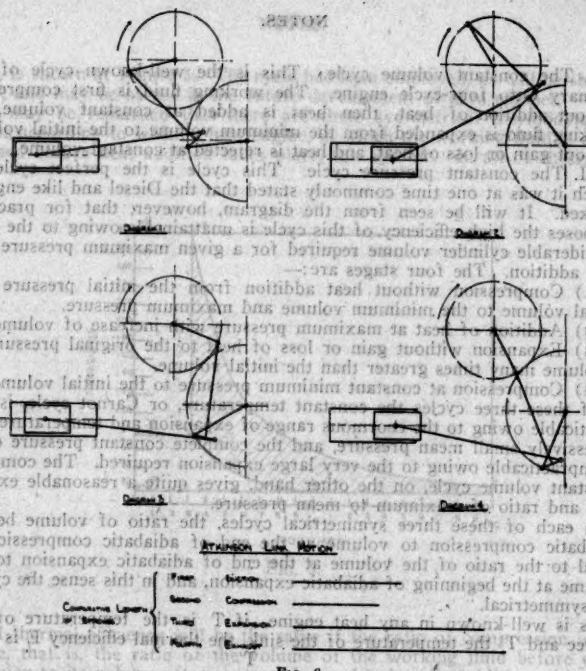
Diagram 1.
Diagram 2.
Diagram 3.
Diagram 4.
Diagram 5.
ATKINSON LIFE ENGINE
Compression
Combustion
Expansion
Exhaust
Suction

FIG. 6.

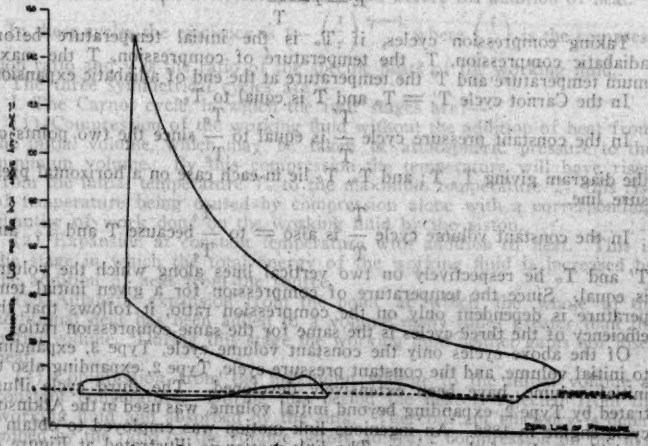

FIG. 7.

B.H.P. at a consumption of 22.6 cubic feet of gas per B.H.P. hour at 131 revolutions per minute, with a mean pressure of about 46 pounds per square inch. The mechanical efficiency was 85 per cent and brake thermal efficiency about 20 per cent. A diagram taken on the Atkinson engine is shown at Figure 7. This cycle is also used in the Humphrey pump.

It should be noted that in all engines working on the constant volume cycle the combustion is not instantaneous, and in many cases combustion continues considerably after the point of maximum pressure is reached. The effect of this is to produce a diagram in which part of the heat may be said to be added at constant volume and part at constant pressure. Attempts have been made by Sir Dugald Clerk and others to produce this condition by regulating the fuel supply, and it has been suggested that the cycle should be termed the dual combustion cycle. The control of combustion would, however, be so uncertain that in considering ideal cycles no very definite air standard cycle could, in the author's view, be evolved.

The Blackstone oil engine would appear to work on this cycle.

It may be noted that by keeping the inlet valve of an ordinary four-stroke cycle engine open during a portion of the compression stroke a diagram is produced which is similar to that obtained in the Atkinson cycle. This is of interest in view of certain recent proposals for limiting the power at ground level of an aero engine specially designed for the low atmospheric pressures at high altitudes.

The internal combustion engine had to go through a long practical development before it reached the stage at which it could be employed for flight purposes.

The first important stage in the development was the use by Otto of the Beau de Rochas cycle in the production of the first compression engines in 1880, and this type has been adopted in practically all aero engines of the present day.

The Otto engine utilized town gas as fuel. The mean pressure was low, only about 70 pounds to the square inch, with an indicated thermal efficiency of 16 per cent and brake efficiency about 13.9 per cent. The cylinder was 171.9 mm. diameter by 340 mm. stroke and the engine ran at a speed of 157 revolutions per minute.

The success of the early Otto gas engines quickly led to the production of heavy oil engines working on the same cycle, but these, owing to their considerable weight, have not yet reached the stage at which they can be employed for aero motors.

At the early period also, engines working on the two-stroke cycle were designed and built by Sir Dugald Clerk and engines of this type were constructed of large powers, chiefly on the Continent. The four-stroke cycle, however, was very much more extensively used and developed.

For light-weight aero motors the two-stroke cycle deserves serious consideration as offering advantages, particularly in connection with direct fuel injection.

A very great advance was made by Gottlieb Daimler using petrol as fuel in engines of comparatively small size, and the work of Daimler led him to the first application of the Otto type of internal combustion engine to motor vehicles. A section of the Daimler engine as applied by Gottlieb Daimler to a motor carriage in 1896 is given at Figure 8. This engine of Daimler may be regarded as the prototype of the modern forms of car, motor boat and aero engines. Engines following this type were the first which were successfully used for flight.

At about the same time as the rapid development in the four-stroke cycle constant volume engine took place, engines were also developed working on the constant pressure cycle. The earliest example of these engines is the

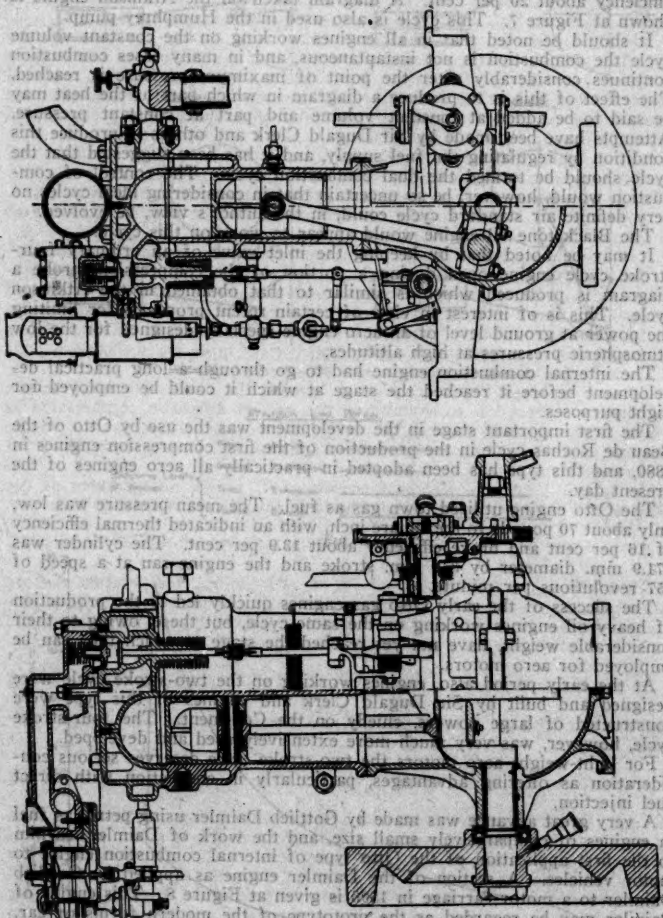

FIG. 8.

FIG. 8.

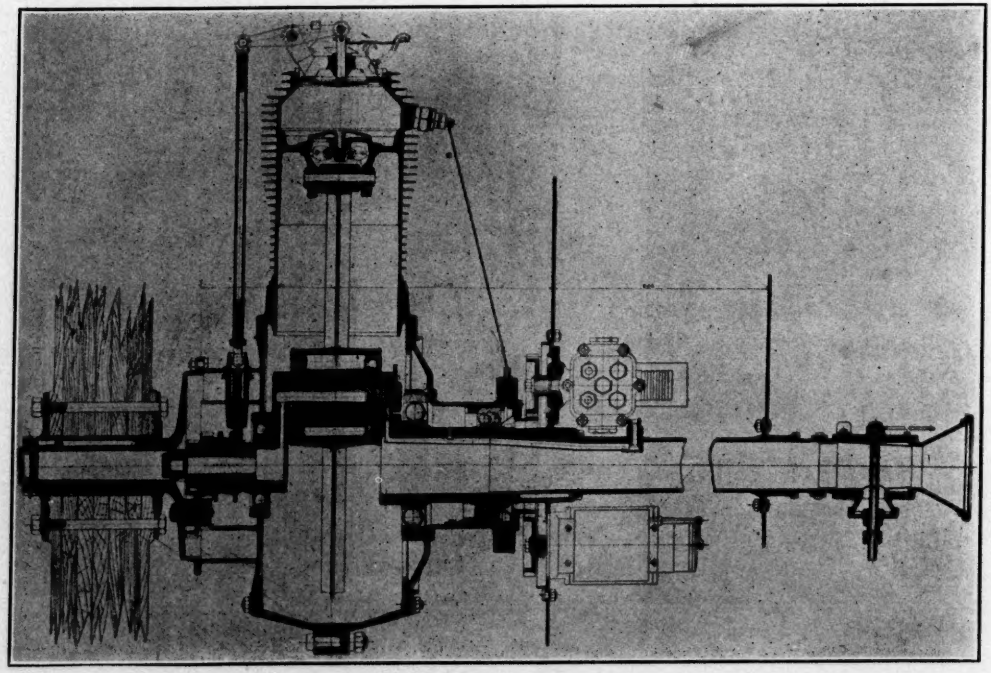

FIG. 9.

Brayton petroleum engine in which the charge of fuel or air was compressed in one cylinder and ignited as it passed into a second or motor cylinder. The constant pressure type has been extensively developed, and was very early proposed for motor vehicles in the U. S. A. The type is represented in modern practice wholly or in part by the Akroyd Stuart, Diesel and Semi-Diesel engines.

The weights of internal combustion engines depend on the compression pressure used, and for the same cycle the weight will have to be greater proportionately with the compression pressure, and cycles involving very high compression ratios or very long expansions and giving increased economy, also increase the weight of the engine; efforts to utilize a longer expansion by compounding the engine have not been successful owing to heavy heat loss in transfer to the low pressure cylinder. For aero engine use, particularly for heavier-than-air machines, weight of engine is of fundamental importance, and it appears that for this purpose the constant volume type of engine is to be preferred. For engines where weight is of less importance and high economy involving less weight of fuel to be carried is the governing factor, engines of the constant pressure type with self-igniting cycles may prove to be more suitable. Such engines may quite well turn out to be extremely suitable for airship work.

The first really successful engine used for flight was that of the Wright Brothers, used in their early flights about 1903. This engine was of the constant volume type with four vertical cylinders 4¼ inches diameter by 4 inches stroke, and developed 24 B.H.P. at a speed of 1,200 revolutions per minute with a petrol consumption of .825 pound per B.H.P. hour, giving a brake thermal efficiency of about 15.5 per cent. The weight of the engine was 210 pounds. The separate cast steel cylinders had inlet valves of the automatic type, and exhaust valves operated by push rods. Thin aluminium water jackets were provided on the cylinder barrels. The weight per B.H.P. was over 7 pounds.

In France, successful flights were made using the Antoinette engine. This was a Vee type with eight cylinders, 4.35 inches bore by 4.15 inches stroke, giving 49 B.H.P. at 1,100 revolutions per minute, and weighing 265 pounds; that is, about 5.4 pounds per horsepower.

In both the Wright and the Antoinette engines the brake mean pressure was about 65 pounds per square inch. The results obtained with these two motors may be said to have clearly demonstrated the suitability of the four-stroke cycle petrol engine to provide motive power for heavier-than-air machines.

A further engine development in France before 1909, and of historical interest, is the three-cylinder radial air-cooled Anzani engine used by Blériot in the first Channel flight. This engine developed 25 B.H.P. at a speed of 1,400 revolutions per minute. The cylinders were of cast iron fitted with valves in a side pocket, the inlet valves being automatic and the exhaust valves operated by push rods. The cylinders were of 3.94 inches bore by 5.92 inches stroke, piston speed 1,380 feet per minute, brake mean effective pressure 65.5 pounds per square inch, and petrol consumption 0.6 pound per B.H.P. hour, giving a thermal efficiency of 21.2 per cent. The weight of the engine was 140 pounds or 5.6 pounds per B.H.P.

An interesting engine for aviation was proposed by Messrs. Burlat in 1904, but did not attain any practical success. In this engine the cylinders and crankshaft rotate in the same direction, the crankshaft speed being twice that of the cylinders. A two-throw crankshaft was employed connected to the centers of two rigid connecting rods, each of which had a piston at each end, the pistons working in opposed cylinders. The crankshaft axis was situated at one-half the radius from the center of the trunnions on which the cylinders revolved, and the length of crank throw was

also one-half this radius. In the engine, as designed, the propeller was fitted to the crankshaft, but the mechanical arrangement suggests that a reduction of propeller speed relative to engine speed can be obtained without the use of gearing.

The most successful engine produced for flight in the early period of development was the Gnome engine designed by M. Seguin. This was the forerunner of the well-known radial air-cooled rotary type which is still very extensively used.

In this type the cylinders are arranged radially and rotate about the axis of a single throw crankshaft which is fixed, and through which the fuel and lubricating oil are fed; thus the pistons rotate about a fixed crank pin. An odd number of cylinders are required in order to obtain working strokes at equal angular intervals during each revolution of the cylinders. In the earliest type, five cylinders were employed of 3.94-inch bore and 3.94-inch stroke; later, the well-known seven-cylinder 80 horsepower type, Figure 9, was produced. These engines were much lighter than any other engine produced at this early date, and were very widely used.

The chankshaft is in two parts, fitted together at the crank pin, the long end being fixed in the engine bearer plates on the aeroplane. The crank case and cylinders rotate about the crankshaft on ball bearings, while the connecting rods are connected to a banjo-shaped big end formed on the master rod. The cylinders are machined from solid nickel chrome steel ingots, the thickness of the barrels being only 1.5 mm. The exhaust valves are situated in the cylinder heads and are actuated by push rods and rockers, while the inlet valves are automatic and are situated in the heads of the pistons.

Fuel and air are led to the engine through the hollow crankshaft from a jet carburetor and the mixture passes from the crank-case to the cylinder through the valves in the pistons.

Although the fuel and oil consumption was heavy, as compared with contemporary stationary engine practice, the very light weight per horsepower, coupled with accessibility and ease of dismounting, made the type easily the most successful aero motor at the date of its first production.

The following table No. 4 gives some particulars as regards the aero engines which were in existence about 1910.

Since 1909 aero engine development has been rapid, and a number of different mechanical types have been produced. The symbolic representation of the main types which have been employed is shown in the diagram, Figure 10.

Of Type I, the stationary vertical engine with cylinders in line, may be mentioned Beardmore, Green, Benz, Mercédès and Maybach, which were all successfully used and had shown reliability before the commencement of the War.

The Beardmore type was constructed for powers of 90 and 120 horsepower before the War, and was very largely used. The engines had six cylinders set slightly on the forward side of the crankshaft axis. Water-cooling was employed, copper water jackets being deposited electrolytically. The crank-case was in two parts, the lower half being removable without disturbing the crankshaft. The valves were of the overhead type operated by push rods and overhead rockers with special laminated springs. The dry weight of the 90 horsepower engine was 5.8 pounds per B.H.P., and of the 120 horsepower engine 5.3 pounds per B.H.P.

The Green engine was also a water-cooled engine with six vertical cylinders.

The Benz 100-horsepower engine obtained the first prize in the German official Aeroplane Trials in 1913, with the remarkably low fuel consump-

TABLE 4.

AERO ENGINES BEFORE 1910

Name of Engine	Wright	Antoinette	Anzani	Green	Wolseley	Pipe	Aster	Renault	Panhard	Gnome
Type of Engine	Vertical	Vee	Fan	Vertical	Vee	Vee	Vertical	Vertical	Vertical	Rotary
Cooling System	Water	Water	Air	Water	Water	Water	Water	Water	Water	Air
Number of Cylinders	4	8	3	4	8	8	4	4	4	7
Bore of Cylinders. Inches	4.25	4.35	3.94	5.52	3.74	3.94	5.12	4.33	7.28	4.35
Stroke of Piston. Inches	4.33	4.15	5.92	5.75	5.0	3.94	5.52	6.30	7.88	4.73
Brake Horse-power	24	48	25	60	54	55	51	38.5	112	45
Brake M.E.P. lbs. sq. in.	64.48	60.52	65.33	78.50	88.47	58.15	88.86	74.71	75.15	65.85
Revs. per Minute	1200	1100	1400	1100	1350	1950	1000	1100	900	1100
Piston Speed. Ft. per Min.	870	760	1380	1015	1125	1280	925	1155	1180	870
Dry Weight of Engine. Lbs	210	265	140	236	300	289	242	286	836	172
Weight per B.H.P. in Lbs.	8.7	5.4	5.6	3.94	5.6	5.3	4.75	7.4	7.5	3.8

NOTES.

tion of 0.47 pound per B.H.P. hour and the low weight of 4.22 pounds per horsepower complete with cooling system less fuel, oil and tanks. The tests at these trials were of a very searching description, and the performance must be regarded as a very excellent one at that date.

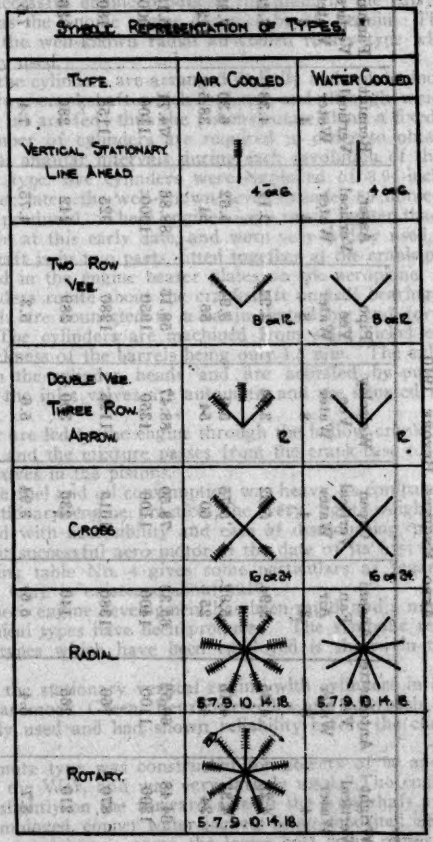

Fig. 10.

The Mercedes early engine developing 88.9 B.H.P., with a fuel consumption of .51 pound per B.H.P. hour and a weight of 4.4 pounds per B.H.P., was a six-cylinder engine, and was awarded the second prize in these trials. This engine embodied many of the characteristic features of the well-known Mercédès construction, which will be dealt with more fully in describing the later engines.

The third prize at the German trials was obtained by the N. A. G. These were four-cylinder engines.

Another interesting German engine is the Inverted Daimler. This obtained the fourth prize in the competition. It developed 65 brake horsepower and the inverted construction gave the advantages of a better view for the pilot, lowered the center of gravity of the engine, and improved the facilities for cooling. It is interesting that no difficulty was found with over-lubrication of the cylinders in spite of their inverted position.

The Maybach engine, as is well-known, was used in the early German Zeppelins, and a similar type was manufactured in this country by the Wolseley Company. An interesting engine of this type was the Argyll 120-horsepower engine, having rotating and reciprocating sleeve valves. This appears to be the only instance in which sleeve valves have been employed on aero engines. This engine competed in the War Office trials in 1914.

Of Type 2, the Vee Type, engines were constructed by the Wolseley Company, Sunbeam, Renault, de Dion, and others. The Wolseley engine was an eight-cylinder water-cooled engine, bore 3.75 inches and stroke 5 inches and developed 54 B.H.P. at a speed of 1,350 revolutions per minute.

Type 3, the three-row engine, is a development of the fan-shaped Anzani type, but was not extensively developed until a later period.

The same is the case with regard to Type 4, the cross engine having rows of 3, 4, 6 or more opposed pairs of cylinders placed at right angles.

With regard to Type 5, Radials, in development of the old Y type Anzani, two interesting engines at the War Office trials were the British Anzani, a radial engine having ten cylinders arranged in two groups of five and air-cooled and the water-cooled nine-cylinder Salmson engine.

The air-cooled radial is of great interest in view of recent developments.

Type 6, the rotary, was, perhaps, the most successful of the earlier aero engines. The earlier Gnome engines and the Monosoupape Gnome nine-cylinder have always been pre-eminent and the Mono up to the present date has been very largely used, latterly chiefly for training purposes. The engine is of bore 110 mm. and stroke 150 mm., and was by far the lightest engine at the British trials, the weight being about 2.7 pounds per B.H.P. The consumption, however, of 0.7 pound per B.H.P. hour is high. The consumption of oil in this engine is also high.

The early Le Rhone engines were also produced about this time. The distinctive features of this type are the cylinder construction with a cast iron liner, connecting rod assembly, and the valve operation. These engines have been very successful and have shown considerable reliability.

An interesting type of engine manufactured in 1914 is the swash-plate type, an example being built for the Statax Engine Company.

In this type five cylinders are arranged parallel with the axis of rotation around a fixed hollow shaft and are held between two casings which rotate with them. The connecting rods are connected to the pistons by universal joints and their big ends are coupled to a ring mounted on ball and thrust bearings on an inclined swash-plate fixed to the shaft. The ring is constrained to rotate with the cylinders and casing by guides on the casing surrounding the swash-plate and thus any point on it rotates about the shaft axis and also oscillates in a rotating plane passing through the shaft axis. The cylinders are cooled by aluminium plates through which they are threaded. The inlet valves are operated directly by cams rotating concentrically with the shaft and the exhaust valves indirectly through push rods and tappets. This engine was claimed to develop 44 B.H.P. with a petrol consumption of 0.53 pound per horsepower hour and to weigh 176 pounds or 4 pounds per horsepower.

The leading particulars of aero engines in 1914 are given in Tables 5 and 6.

Table 5.

Particulars of Leading Engines in 1914. Test Figures.

British War Office Trials, 1914.

Make of Engine		Argyll	British Anzani	Beardmore Austro-Daimler		Dudbridge Salmson Canton-Unne			British Gnome Monosoupape		Green		Sunbeam	Wolseley.	
														F2	Renlt.
Type		Vertl.	Radl.	Vertl.	Vertl.	Radl.	Radl.	Radl.	Roty	Roty.	Vertl.	Vertl.	Vee	Vee	Vee
Number of Cylinders	No.	6	10	6	6	7	9	14	9	9	6	6	8	8	12
Bore of Cylinders	Ins.	4.92	4.53	4.72	5.12	4.72	4.72	4.72	4.33	4.13	5.51	5.51	3.54	5.00	3.78
Stroke of Pistons	Ins.	6.89	6.10	5.51	6.89	5.51	5.51	5.51	5.90	5.90	5.98	5.98	5.90	7.00	5.51
Brake Horse-Power	B.H.P.	116.6	100.2	95.5	129.0	89.0	124.0	185.0	99.2	104.2	101.7	103.5	130.9	139.6	113.5
Brake Mean Pressure	lbs sq"	48.95	36.39	50.99	47.12	40.89	44.25	41.45	42.71	48.00	37.72	38.52	54.37	39.44	33.64
Revs. per Minute	R.P.M.	1200	1110	1279	1275	1275	1276	1307	1206	1205	1246	1242	2046	1238	1800
Piston Speed	Ft Mn.	1378	1129	1175	1464	1171	1172	1200	1187	1186	1243	1239	2014	1445	1654
Fuel per B.H.P. Hour	Lb.	0.57	0.95	0.69	0.54	0.58	0.55	0.61	0.69	0.72	0.67	0.63	0.53	0.58	0.66
Weight of Engine complete with Cooling System	Lbs.	600.0	464.0	437.1	583.8	432.9	545.6	780.2	275.0	275.0	506.1	507.3	565.1	810.7	478.8
Weight per B.H.P.	Lbs.	5.15	4.64	4.68	4.52	4.86	4.40	4.22	2.77	2.64	4.97	4.90	4.32	5.81	4.22
Weight of Power Plant with Fuel, Oil and Tanks for 4 hrs. ..	Lbs.	919.2	958.7	738.1	904.6	671.9	871.5	1303.9	655.4	716.0	835.2	818.2	896.1	1190.6	863.2
Weight of Power Plant, etc., per B.H.P.	Lbs.	7.9	9.5	7.7	6.96	7.55	7.20	6.95	6.60	6.86	8.22	7.91	6.84	8.54	7.60

Table 6.

Particulars of Leading Engines in 1914. Test Figures.

German Aeroplane Engine Trials, 1913.

Make of Engine		Benz.	Mercedes—Daimler							Argus		N.A.G.			Mu-lag.	Schro-ter	Bayer-ischen	Gyro
Type		Vertl.	Vertl.	Vertl.	Vertl.	Vertl.	Vertl.	Vertl.	Vertl.	Vertl.	Vertl.	Vertl.	Vertl.	Vertl.	Vertl.	Roty.	Roty.	
Number of Cylinders	No.	4	6	4	4	4	6	4	4	4	4	6	4	4	6	6	7	7
Bore of Cylinders	Ins.	5.12	4.13	4.72	5.51	4.72	4.72	4.33	4.72	5.51	4.92	4.92	5.31	4.72	4.33	4.88	4.33	4.29
Stroke of Pistons	Ins.	7.09	5.51	5.51	5.90	5.51	5.51	5.51	5.51	5.51	5.12	5.12	6.30	4.72	6.69	6.30	4.72	4.76
Brake Horse-Power	B.H.P.	101.2	88.9	71.3	99.1	70.4	103.0	60.8	66.3	96.6	71.0	102.0	95.6	95.8	101.7	88.7	37.4	38.7
Brake Mean Pressure	Lbs sq"	107.0	114.2	103.5	102.0	107.5	107.0	104.8	98.0	106.4	107.5	101.1	101.0	95.0	101.1	79.2	66.6	77.0
Revs. per Minute	R.P.M.	1288	1387	1412	1373	1343	1315	1369	1391	1368	1342	1370	1344	1408	1346	1252	1031	954
Piston Speed	Ft. Min.	1523	1274	1297	1350	1233	1207	1282	1278	1256	1145	1169	1109	1400	1557	1314	812	757
Fuel per B.H.P. Hour	Lb.	0.47	0.51	0.50	0.49	0.50	0.53	0.50	0.50	0.53	0.59	0.59	0.52	0.49	0.53	0.63	0.86	0.77
Weight of Engine Complete with Cooling System	Lbs.	425	390	352	490	370	541	337	382	427	357	536	279	476	588	468	177	186
Weight per B.H.P.	Lbs.	4.20	4.37	4.91	4.91	5.36	5.23	5.51	5.72	4.40	5.00	5.22	4.96	4.98	5.76	5.27	4.72	4.80
Weight of Power Plant with Fuel, Oil and Tanks for 4 hrs.	Lbs.	857	605	525	719	545	799	502	546	692	566	829	409	690	834	705	340	351
Weight of Power Plant per B.H.P.	Lbs.	4.48	6.81	7.37	7.25	7.74	8.21	8.26	8.22	7.16	7.98	8.12	7.33	7.28	8.21	7.94	9.10	9.07

Lecture II.—Delivered January 24th, 1921.

When war was declared, among the engines of which immediate use was made, were the following:—

British—R.A.F., Beardmore, Gnome, Le Rhone, Renault, Curtiss, Green.
French—Gnome, Le Rhone, Renault, Anzani, Canton Unné.
German—Mercédès, Benz, and Argus, German Gnome, Oberursel; and for airship purposes the Maybach.

The average B.H.P. of the engines available at that date was about 100; the average B.M.E.P. was 71 pounds square inch, with a piston speed of 1,280 feet minute, and a weight per horsepower of 7.7 pounds.

Of these engines, the Beardmore, Green, Mercédès, Benz, Argus and Maybach were of type I, that is, vertical cylinders in line; the R.A.F. and Renault were air-cooled, and the Curtiss water-cooled Vee type stationary engines; the Gnome and Le Rhone were rotary engines; the Anzani an air-cooled radial, and the Canton Unné, a water-cooled radial. It is seen that while the British had very few engines available, they immediately adopted certain existing French types, some of which were subsequently built in large quantities in Great Britain.

Of the French engines available during the early period of the war, the rotary type was perhaps the most used throughout, though the development, except in power, was not striking. The Vee type was largely used, and was very successfully developed as a water-cooled engine by various manufacturers, notably the Hispano type, designed by M. Birkigt. The essential characteristics of this engine are the high rotational speed and the simple valve gear; this type was produced in very large quantities in England, France, America, Spain and Italy. The Renault 12-cylinder water-cooled was another engine of the same type.

The Germans adopted for the very large bulk of their work, a single type on which there was available a large amount of data and experience built up in connection with racing engines for motor cars, boats and other purposes. They consistently kept to this type throughout the whole period of the war, except in the case of a few rotary engines, which are copies of designs got out originally by the French. At a late stage they were also attempting to bring the high speed 8 and 12 cylinder Vee type into production. For example, the Benz Company were building these engines following the Hispano design with gearing of the Rolls Royce type.

The characteristic of the German engines used in the war is the attempt to obtain high fuel economy and reliability with less consideration of the feature of light weight per horsepower than had been given by the Allies.

Many different types as to which little experience was available, were tried by the Allies throughout the war, and in order to get lighter weight, higher stressing was allowed than in German practice, with perhaps some sacrifice of reliability at the outset. Many types were but partially developed and were subjected to somewhat severe criticism before the development had been carried far enough for definite results to be obtained to allow the type to become standardized.

It may be stated that in Germany only the stationary vertical type of engine was seriously developed, and in France, only the Vee water-cooled type, both being developments of standard motor car engine practice.

In Great Britain development took place in several types, mainly the straight six as the B.H.P. Puma; the Rolls Royce Vee, and the air-cooled rotary. Large quantities of each of these types were produced.

The general course adopted with all types during the war was to increase the power by using higher compression and higher speed, overcoming the mechanical difficulties arising from these changes as and when they

arose. Thus the Rolls Royce Eagle, which was originally rated at 255 horsepower, was ultimately boosted up to 375 horsepower. Similar procedure was adopted in Germany, but on more conservative lines.

This boosting up of the various types of engines was accompanied by considerable improvement in the materials employed in construction; for example the strength of steel used was increased from 40 tons to values of the order of 100 tons and very great advance was made in the quality of aluminium alloys enabling this material to displace iron and steel with consequent advantages as regards heat dissipation and saving in weight.

The most important original developments of the purely aero engine since 1914 have taken place in this country because we experimented with nearly all types. Progress was made particularly in the development of various air-cooled types. The Germans kept throughout to the water-cooled engines; the British used both air and water-cooling in different types from the beginning; first air with the R.A.F. engine, then water; though the air-cooled engine with its manifestly considerable advantages for military purposes, was, after being closely investigated as a scientific problem, perhaps more favored as a type at the close of the war.

The Anzani, one of the very earliest of the fixed radial engines, was used at the beginning of the war, but this type was not developed to any great extent until 1918, and even then the results obtained hardly came up to expectations.

Again, the rotary engine, built in very large quantities throughout the whole period of the war, showed little real advance as regards design, although the power was considerably increased; such points, for example, as effective cooling of the following side of the cylinder being very incompletely explored; while the design of the cylinder head was always deficient in the provision of adequate heat-dissipating surface. The backward state of development of these types may perhaps be explained by the fact that the majority of aero engine manufacturers had gained their experience in car engine practice and would naturally select for development the type of engine with which they were best acquainted, or a close variation of this type, rather than select a new and different type, such, for example, as the air-cooled radial.

It will be interesting to follow the development of these air-cooled radial and rotary types when the demand for an aero motor of a weight per horsepower not attainable with other designs is brought to the serious attention of our best technical men.

In order to study the progress in the different types through the war period, the engines may be grouped under certain standard types.

Six different types are diagrammatically represented in Figure 10 ("Journal," Vol. LXIX, p. 701). It is of interest to describe some of the best known engines of each of these types.

Taking first the vertical or I. type, Table 7 shows the particulars of the many German engines, and Table 8 the few British engines of this type produced during the war.

At the beginning this type was represented in Great Britain by the 120-horsepower Beardmore and in Germany by the 100-horsepower Mercédès, the 120-horsepower Argus, and engines of 160 horsepower manufactured by the Mercédès and Benz Companies.

The 120-horsepower Beardmore was early followed by a more powerful type developing 160 B.H.P., and the latter engine was used with great success and to a very large extent during the war.

The 160-horsepower engine has six separate cylinders in line with electrolytically deposited copper water jackets. The crankshaft is placed slightly in advance of the central plane of the cylinders, and the lower half of the crankcase is removable without disturbing the crankshaft bearing

TABLE 7.

GERMAN VERTICAL WATER COOLED ENGINES.

Make of Engine.		Mercédès	Argus	Benz	Mercédès	Mercédès	Argus (Opel)	Mercédès	Austro Daimler	Benz	Bayern	Mercédès	Mercédès	Basse Selve	Maybach
Number of Cylinders	No.	6	6	6	6	6	6	6	6	6	6	8	6	6	6
Bore of Cylinders	Ins.	4.72	5.12	5.12	5.51	5.51	5.71	5.51	5.31	5.71	5.90	5.51	6.30	6.10	6.50
Stroke of Pistons	Ins.	5.51	5.51	7.09	6.30	6.30	6.30	6.30	6.89	7.48	7.09	6.30	7.09	7.87	7.09
Total Swept Volume of Engine	Cu.Ins	579.5	681.0	876.3	901.2	901.2	967.4	901.2	916.8	1149.0	1164.6	1202.2	1326.0	1381.2	1412.0
Compression Ratio	R.	4.6	4.65	4.5	4.5	4.64	4.75	5.73	5.02	4.95	6.42	4.73	4.94	4.34	5.95
Brake Horse Power	Nml.	102.0	121.5	160.0	162.5	174.0	180.0	204.0	200.0	230.0	234.0	248.0	252.6	*269.0	294.0
Revolutions per minute ..	Nml.	1300	1300	1400	1400	1400	1400	1600	1400	1400	1400	1400	1400	1400	1400
Piston Speed in Feet per min.	Ft.Mn.	1195	1195	1655	1470	1470	1470	1680	1607	1745	1655	1470	1655	1837	1655
Brake Mean Pressure	lbs sq"	107.1	108.7	103.0	102.0	109.1	105.0	112.0	123.3	113.2	113.5	116.6	107.5	*110.0	117.7
Fuel per B.H.P. Hour	Lbs.	0.522	0.510	0.520	0.522	0.494	0.594	—	0.499	0.585	0.420	0.524	0.543	—	0.473
Oil per B.H.P. Hour	Lbs.	0.035	0.035	0.31	0.035	0.047	0.018	0.032	0.039	0.025	0.41	0.047	0.036	—	0.041
Brake Thermal Efficiency (18600 B.T.U. per lb.) ..	%	26.20	26.85	26.32	26.20	27.70	23.00	—	27.40	23.40	32.75	26.10	25.20	—	28.90
Weight of Engine complete with Cooling System	Lbs.	511.3	599.0	710.0	738.6	773.1	885 0	797.7	858.5	1012.8	795.5	1061.2	1126.0	1060.0	1102.0
Weight per B.H.P.	Lbs.	5.01	4.93	4.44	4.55	4.44	4.92	3.91	4.29	4.40	3.40	4.28	4.47	3.94	3.75

NOTE.—Figures marked with an asterisk are based on estimated horse-power. Engine was damaged so that tests could not be made.

caps. The construction of the crankcase in this respect differs essentially from that very largely adopted in the German engines of this type. The cylinders are of cast iron with steel flanges screwed on to the base, and are separately bolted to the crankcase. The valves are of the overhead type operated by push rods and overhead rockers, and are held to their seats by pivoted leaf springs, each end of a spring controlling an inlet and exhaust valve respectively. The inlet valve and cage can be removed by merely unscrewing the inlet bend.

In the earlier German engines, the Mercédès, Argus and Benz, the crankshaft journal bearing caps are cast in one with the lower half of the crankcase. By this means considerable stiffness is given to the crankcase, and to the engine as a whole, and the construction is characteristic of nearly all the German engines. It is a useful feature in engines for aeroplanes in which there is no rigid bed provided for the engine.

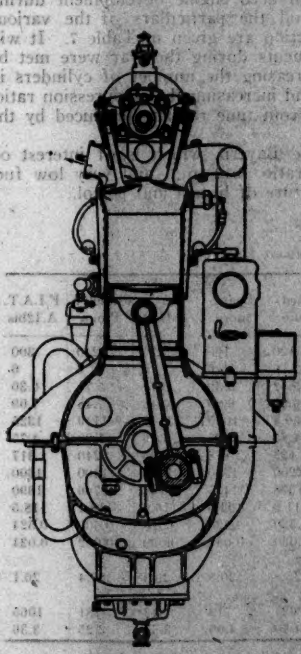

FIG. 11.

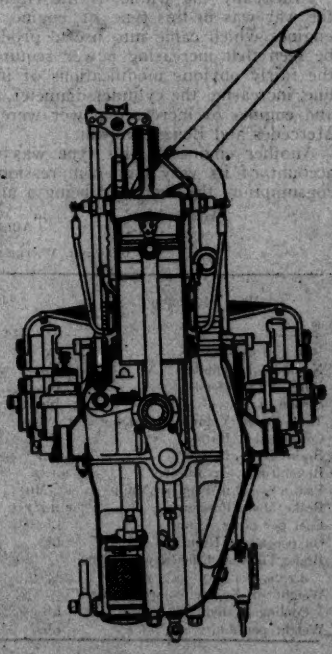

FIG. 12.

The best known of the earlier engines mentioned are the 160 Mercédès and the 160 Benz, figures 11 and 12. As these engines are typical of two main German types it will be interesting to compare them. Both adopt the typical German crankcase construction referred to above.

In the Mercédès type, overhead valves are employed, actuated by an overhead camshaft situated over the tops of the cylinders and operating the valves through rocker levers, a construction which, applied to the Vee type, is also found in the Rolls Royce and Liberty Engines.

In the Benz construction, the camshaft is housed in the upper half of the crankcase, and the valves are actuated by push rods and rocker levers.

The Maybach engine which was largely used in airships from the very beginning of the war employed the long push-rod type of valve gear. It was always particularly well-made and the size was increased from under 200 horsepower to 300 horsepower; the latter size was also constructed as an aeroplane engine.

In the Argus construction, the valve gear is of the Benz type. The later engines, with the exception of the Maybach, all follow either the Mercédès or the Benz general design. This variation of the valve gear is the main difference in German types.

Practically the whole of the German aero engine development during the war was in this type of engine, and the particulars of the various engines which came into useful production are given in Table 7. It will be seen that increasing power requirements during the war were met by the fairly obvious modifications of increasing the number of cylinders in line, increasing the cylinder diameter, and increasing the compression ratio, and engines of increased power were from time to time produced by the Mercédès and Benz Companies.

Another engine of this type was the Bayern, which is of interest on account of its very high compression ratio, 6.42 to 1, and very low fuel consumption, the fuel used being a mixture of benzol and petrol.

TABLE 8.

VERTICAL WATER-COOLED ENGINES.

Make of Engine.		Beardmore.	Beardmore.	Galloway. B.H.P.	Siddeley Puma	F.I.A.T. A.12bis
Rated Horse-Power	H.P.	120	160	230	240	300
Number of Cylinders	No.	6	6	6	6	6
Bore of Cylinders	Ins.	5.12	5.59	5.71	5.71	6.30
Stroke of Pistons	Ins.	6.89	6.89	7.48	7.48	7.09
Total Swept Volume of Engine	Cu. Ins.	850.4	1015	1149	1149	1325
Compression Ratio	R.	4.85	4.56	4.96	5.00	4.75
Brake Horse Power	B.H.P.	133	178	236	240	317
Revolutions per Minute	R.P.M.	1200	1350	1400	1400	1600
Piston Speed in Feet per Minute	Ft./Mn.	1378	1435	1746	1746	1890
Brake Mean Pressure	B.M.E.P.	111.0	103.0	116.0	118.1	118.5
Fuel per B.H.P. Hour	Lbs.	0.530	0.520	0.520	0.500	0.524
Oil per B.H.P. Hour	Lbs.	0.030	0.032	0.030	0.062	0.031
Brake Thermal Efficiency. (18600 B.T.Us. per lb.)	%	25.8	26.3	26.3	27.4	26.1
Weight of Engine complete with Cooling System	Lbs.	624	713	843	781	1065
Weight per B.H.P.	Lbs.	4.70	4.00	3.57	2.25	3.36

NOTE.—A special high compression Puma Engine has developed 300 B.H.P. at 1650 r.p.m. with a B.M.E.P. of 130lbs. per square inch.

The development of the stationary vertical engine in Great Britain resulted in the production of engines of about the same power as the German engine, but of a lower weight per horsepower. The leading particulars of the chief engines of this type produced are given in Table 8. From this table it will be seen that the B.H.P. developed into the Puma, which was

in production in its latest state of development at the time of the Armistice, and gave 240 B.H.P. at 1,400 revolutions per minute with a B.M.E.P. of 118 pounds per square inch, and a compression ratio of 5 to 1. The weight was only 3.25 pounds per B.H.P. as compared with the average weight of the larger German engines, namely 4.05 pounds per B.H.P. An engine of the Puma type with compression ratio raised to 5.4 developed on test 308 B.H.P. at 1,650 revolutions per minute, with a consumption of

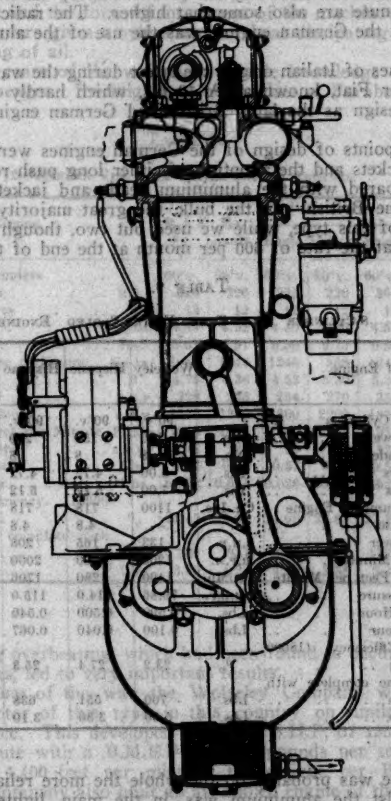

FIG. 13.

0.497 pounds of petrol per B.H.P. hour. The weight was 2.51 pounds per B.H.P. This high compression engine did not, however, come into production.

In the Puma engine, Figure 13, the cylinders are arranged in blocks of three, each block being provided with a common water jacket. The bore and stroke are 5.71 inches and 7.48 inches, respectively. The petrol con-

sumption at ground level is 0.5 pound per B.H.P. hour and the oil consumption 0.062 pound per B.H.P. hour. The cylinder head casting is made of aluminum alloy. Two exhaust valves and one inlet valve are provided situated vertically in the cylinder crown and the bronze valve seats are screwed and shrunk into the cylinder heads, as are also the steel cylinder liners. The engine is developed from the earlier Galloway B.H.P., the chief feature of difference being the aluminium heads instead of cast iron which was used in the Galloway B.H.P. The compression ratio and revolutions per minute are also somewhat higher. The radical difference between this and the German engines was the use of the aluminium cylinder head.

Of the engines of Italian origin, the latest during the war period was the 300-horsepower Fiat, known as A 12 bis, which hardly shows the same progress in design as the later British and German engines of the same type.

The salient points of design of the German engines were steel cylinders and welded jackets and the adoption of either long push rods or overhead camshaft compared with the aluminium heads and jackets and overhead camshaft of the British. In the bulk, the great majority of all German engines were of this type, while we used but two, though one, the Puma, was produced at the rate of 600 per month at the end of the war.

TABLE 9.

8-CYLINDER VEE TYPE WATER-COOLED ENGINES.

Make of Engine		Wolseley	Hispano	Hispano	Hispano	Wolseley Viper
Arrangement of Cylinders		90°v.	90°v.	90°v.	90°v.	90°v.
Rated Horse Power	B.H.P.	130	150	200	300	180
Number of Cylinders	No.	8	8	8	8	8
Bore of Cylinders	Ins.	5.00	4.72	4.72	5.51	4.72
Stroke of Pistons	Ins.	7.00	5.12	5.12	5.91	5.12
Total Swept Volume of Engine	Cu. Ins.	1100	718	718	1127	718
Compression Ratio	R.	—	4.8	4.8	5.34	5.30
Brake Horse Power	B.H.P.	133	165	208	300	220
Revolutions per Minute	R.P.M.	1200	1600	2000	1800	2000
Piston Speed in Feet per Minute	Ft./Mn.	1400	1280	1706	1770	1706
Brake Mean Pressure	B.M.E.P.	79.86	114.0	115.0	116.8	121.8
Fuel per B.H.P. Hour	Lbs.	0.590	0.500	0.540	0.576	0.520
Oil per B.H.P. Hour	Lbs.	0.100	0.040	0.067	0.040	0.050
Brake Thermal Efficiency. (18000 B.T.Us. per lb.)	%	23.2	27.4	25.3	23.75	26.3
Weight of Engine complete with Cooling System	Lbs.	700	551	638	795	657
Weight per B.H.P.	Lbs.	5.30	3.30	3.10	2.60	3.10

The steel type was probably on the whole the more reliable in quantity production whilst the aluminium was, in the main, lighter. Thermodynamically the German engines appeared to be somewhat superior, due to the development of one type with larger cylinders. The German engines had also to undergo severe tests before acceptance.

For engines of higher powers, it would appear that the Allies preferred to go to other types, especially the water-cooled Vee; and it may be said that the most serious development of engines for production both in this country and in France was in the latter type.

Particulars of this type are shown in Tables 9, 10, and 11. Although at the beginning of the war some of the engines such as the Wolseley-Renault and R.A.F. engines were air-cooled, the more recent examples have all been of the water-cooled type. The arrangement of the cylinders of Vee engines is not well adapted to the provision of adequate wind speed over all parts of the cylinder, and in recent practice air-cooling has been mostly confined to engines of the rotary and static radial types in which one set of cylinders is not screened by another from the wind, due to propeller slip stream and the forward movement of the machine: this arrangement of the cylinders gives ample space for heat dissipating fins, ensuring more even cooling of all.

In the earlier years of the war the available knowledge of air-cooling was somewhat meagre. Investigations undertaken at Farnborough, with a

TABLE 10.

12-CYLINDER VEE TYPE WATER-COOLED ENGINES.

Make of Engine		R.A.F 3a.	Renault	Rolls-Royce. Eagle	Falcon	Eagle	Liberty 12a.	Sunbeam Maori
Arrangement of Cylinders		60°v.	50°v.	60°v.	60°v.	60°v.	45°v.	60°v.
Rated Horse Power	B.P.	220	220	225	220	360	400	250
Number of Cylinders	No.	12	12	12	12	12	12	12
Bore of Cylinders	Ins.	4.49	4.92	4.50	4.00	4.50	5.50	3.94
Stroke of Pistons	Ins.	5.51	5.91	6.50	5.75	6.50	7.00	5.31
Total Swept Volume of Engine	Cu.In.	1047	1348	1240	868	1240	1650	776
Compression Ratio	R.	4.75	4.36	4.53	5.30	5.30	5.30	5.60
Brake Horse-Power	B.H.P.	220	225	254	270	350	405	265
Revolutions per Minute	R.P.M.	1700	1300	1800	2200	1800	1650	2100
Piston Speed in Feet per Minute	Ft.Mn.	1563	1081	1950	2108	1950	1925	1860
Brake Mean Pressure	B.M.E.P.	98	100	90	114	124	118	125
Fuel per B.H.P. Hour	Lbs.	0.56	0.53	0.57	0.53	0.50	0.49	0.51
Oil per B.H.P. Hour	Lbs.	0.022	0.061	0.026	0.029	0.028	0.032	0.026
Brake Thermal Efficiency. (18600 B.T.U. per lb.)	%	24.43	25.81	24.09	25.81	27.36	27.03	26.83
Weight of Engine complete with Cooling System	Lbs.	938	951	1065	912	1177	1088	1065
Weight per B.H.P.	Lbs.	3.60	4.20	4.20	3.38	3.37	2.69	3.95

view to avoiding overheating, which had been found to occur in the R.A.F. 1*a* and 4*a* engines, led to very important results.

At the beginning of the war the Wolseley Company were building a water-cooled motor of this type in this country on similar lines to the Renault in France. This developed a normal B.H.P. of 133 at 1,200 revolutions per minute with a B.M.E.P. of 79.9 pounds per square inch and piston speed of 1,400 feet per minute; the weight per B.H.P. was 5.3 pounds. This Company also built an air-cooled engine of the Renault type, the largest of which at this date gave 80 B.H.P. The engine most in the public eye in the early days of the war was the 8-cylinder R.A.F. 1*a*; an air-cooled design first built at Farnborough, though afterwards built by many contractors. This engine had cast iron cylinders with valves in side pockets; the cylinders were staggered so that independent connecting rods could be employed, a construction which was not followed in later engines. The pistons were of cast iron with cast iron rings and the crankcase of aluminium. The propeller shaft was driven at half-engine speed by spur

gearing, which also drove the camshaft which was co-axial with it. Only a single thrust bearing was employed so that the engine could only be used as a tractor. The inlet valves were directly operated while the exhaust valves were actuated by rocker levers and push rods from a camshaft with cams cut solid with the shaft housed in the crankcase between the rows of cylinders. Lubrication was by gravity and splash. This engine had roller bearings to its crankshaft and this was probably its most novel feature. It was at the last a very reliable engine, but always too low in power for the work on which it was employed.

TABLE 11.

AIR-COOLED, VEE TYPE ENGINES.

Make of Engine.		Wolseley Renault	R.A.F. 1A.	R.A.F. 4A.	R.A.F. 4D.
Arrangement of Cylinders		90°v.	90°v.	60°v.	60°v.
Rated Horse-Power	H.P.	80	100	150	180
Number of Cylinders	No.	8	8	12	12
Bore of Cylinders	Ins.	4.13	3.94	3.94	3.94
Stroke of Pistons	Ins.	5.12	5.51	5.51	5.51
Total Swept Volume of Engine	Cu. Ins.	549.5	536.8	805.2	805.2
Compression Ratio	R.	4.16	4.20	4.2	4.7
Brake Horse Power	B.H.P.	102	90	160	196
Revolutions per Minute	R.P.M.	1800	1600	1800	1800
Piston Speed in Feet per Minute	Ft./Mn.	1538	1470	1654	1654
Brake Mean Pressure	B.M.E.P.	82.0	83	87.7	107.0
Fuel per B.H.P. Hour	Lbs.	0.660	0.700	0.675	0.532
Oil per B.H.P. Hour	Lbs.	0.055	0.056	0.067	0.054
Brake Thermal Efficiency (18600 B.T.U. per lb.)	%	21.05	19.55	20.27	25.72
Weight of Engine complete	Lbs.	500	468	637	670
Weight per B.H.P.	Lbs.	4.90	5.20	3.98	3.42

The Hispano Companies have been most prominent in the development of the eight-cylinder type, and very large numbers of engines of 200 horse-power of the Hispano type were built by many companies in France and Great Britain. The latest form of the Hispano, the 300-horsepower engine, is the most compact engine of its type and power yet produced. This engine develops 300 B.H.P. with a B.M.E.P. of 116.8 at 1,800 revolutions per minute, piston speed 1,770 feet per minute, and weight 2.60 pounds per B.H.P. The very high speeds of these engines necessitate the use of gearing in order to obtain practicable propeller efficiency, except in very fast scouting machines.

In this type, the cylinders were of steel screwed into a monobloc aluminium water jacket, the inlet and exhaust ports being arranged in the aluminium casting. Tubular connecting rods are used, one of each pair having a forked big end which runs on the outer surface of the other. Overhead valves are used and are operated by an overhead camshaft, the cams acting directly on the valve stems, a form of valve gear particularly suitable for a high speed engine.

The Sunbeam Company also have built engines of this type which were extensively used. The earliest of these, with eight cylinders of 90 mm. bore, and 150 mm. stroke, was fitted with valves in side pockets, and developed 145 B.H.P. at 2,100 revolutions per minute, and weight per horse-power 4.3 pounds. An engine of the same type with slightly larger stroke,

160 mm., developed 170 B.H.P. at 2,000 revolutions per minute, with weight per horsepower 3.75 pounds. Later an overhead valve type was produced with cylinder 120 mm. bore and 130 mm. stroke, giving 200 B.H.P. at 2,100 revolutions per minute, with a weight per horsepower of 3.16 pounds. In all of these, gearing was employed to reduce the propeller speed to about 1,000 revolutions per minute.

The twelve-cylinder Vee engines were very largely used by the Allies, and in this country underwent extensive development, particularly marked in the engines produced by the Rolls Royce Company, two of which were employed in the successful Vickers Vimy Atlantic flight. The steady increase in power and reliability obtained by this firm in their series of engines by means of improvements in the design, proved by experience to be desirable, is very remarkable and would seem to show that important results could be obtained by the application of similar methods and high technical ability to engines of types which have undoubtedly inherent advantages, but are still in a comparatively backward state of development.

The Rolls Royce series of Eagle and Falcon engines, 12 cylinder 60 degrees Vee, have reached very great reliability. In the Eagle series the bore is 4.5 inches and stroke 6.5 inches; in the Falcon 4.0 inches and 5.75 inches, respectively. The cylinder construction and valve operation are similar to the Mercédès. The connecting rods are of the articulated type. The high speed of the engines necessitates reduction gearing.

An early example of the 12-cylinder 60-degree Vee water-cooled type is the R.A.F. 3*a* 220-horsepower. The cylinders with their heads and valve ports are machined from solid steel forgings, with sheet steel water jackets welded in position. The inlet and exhaust valves are in the cylinder head, and are actuated by a single rocker and push rod from a camshaft lying along the Vee, co-axial with and driven by dogs from the propeller shaft. The crankshaft runs in roller bearings, and drives the propeller shaft at half speed through spur gearing. The connecting rods are of the articulated type, and aluminium pistons with cast iron rings are employed.

The R.A.F. 4*a* was largely used throughout the war. Air-cooling is employed, the cylinders being of cast iron with integral cooling fins, and valves in side pockets formed on the inner sides of the cylinders. The engine is developed from the R.A.F. 1*a* eight-cylinder, and is of very similar construction.

The 240-horsepower Renault is a French water-cooled engine, with 12 cylinders inclined at 80 degrees; bore 124 mm., stroke 150 mm., and a normal speed of 1,200 revolutions per minute. Little development of this type took place in France.

The Liberty, Fig. 14, produced in large quantities in the United States, is a 12-cylinder 45-degree Vee; the cylinder construction closely follows the Mercédès and Rolls Royce designs; overhead valves and camshaft of a similar type are employed. The crankshaft journal bearing caps are cast in one with the lower half of the crankcase. The connecting rods are of the direct bearing type, one of each pair being forked at the big end to allow both to bear directly on a phosphor bronze shell on the crank pin. In the design of this engine, special attention was given to ease of production, and ultimately considerable success was achieved.

The Sunbeam Company also produced a number of 12-cylinder water-cooled Vee engines, the best known of which is the Maori, used in the airship R34 in the Atlantic crossing. The two banks of cylinders are inclined at 60 degrees to each other, and are each provided with four overhead valves actuated by two camshafts driven through trains of spur gearing; the connecting rods are of very light H section. As weight is of secondary importance in airship work, a fly-wheel is fitted to the crankshaft.

A governor is fitted, which cuts out the ignition when the engine speed reaches 2,500 revolutions per minute, or when the oil pressure falls below 20 pounds per square inch.

It will be seen that of water-cooled Vee types we had two classes; steel cylinders with steel jackets welded on, and aluminium jackets with steel liners. In practice, the former, used in the Rolls engines proved probably more reliable and the latter method the lighter. The latter was used in

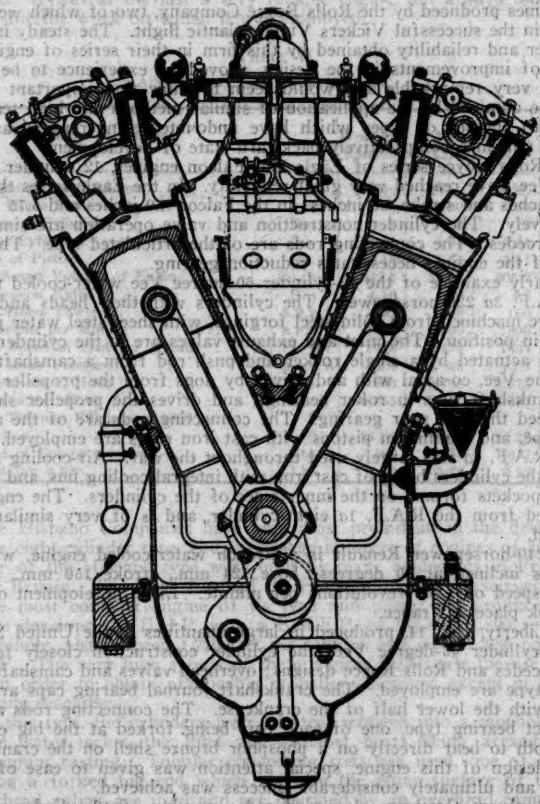

FIG. 14.

high speed scouts and the former in longer formation flights and larger machines. Both types used overhead valve gear, in the latter the cams operating direct on the valve spindles.

In the air-cooled engines of this type the cast iron cylinders kept the weight high; the most novel point was the use of roller bearings for the crankshaft.

As regards the fan or three-row type, little progress was made for some time after the original three-cylinder Anzani and R.E.P. The first engine of high power of this type produced was the Sunbeam 18-cylinder 450-horsepower Viking, with three rows each of six water-cooled cylinders, the angles between the rows being 40 degrees. The bore and stroke are 110 mm. and 160 mm., and the cylinders are cast in blocks of three, with two inlet and two exhaust valves per cylinder operated by two camshafts on each row. Master connecting rods are fitted for the central row of cylinders, and articulated rods for the side rows. Six carburetors are employed, each supplying a block of three cylinders. Each camshaft of the central row of cylinders operates the exhaust valves of one block of the central row and the inlet valves of the other block, the exhaust valves being on opposite sides on the two central cylinder blocks.

The latest development of the three-row type is the Napier Lion, Fig. 15, which has achieved very considerable success since the war. This engine was originally designed to work at full throttle at a height of 10,000 feet only, and to give not less than 300 horsepower at that altitude. As actually built, these engines are capable of running at full power at ground level. The engine has three rows at 60 degrees of four separate cylinders of 5½-

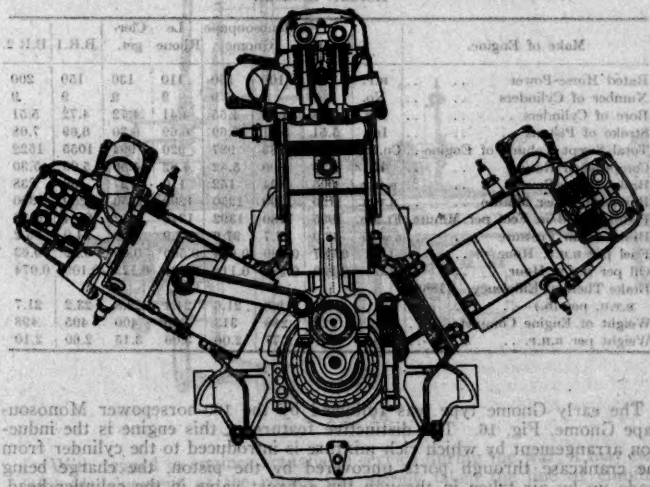

Fig. 15.

inch bore by 6⅜-inch stroke, with a compression ratio of 5.55, and normal B.H.P. 450 at 2,000 revolutions per minute with B.M.E.P. 122 pounds per square inch. The weight is 2.56 pounds per B.H.P. Fuel consumption is .504 pound per B.H.P. hour, and oil consumption .022 pound per B.H.P. hour. This form of construction enables a short engine of high power to be built. The engine is of interest in that it uses a high compression ratio, and in that the design is of the oversized type, the cylinders being designed to work at reduced atmospheric pressure with other parts proportionally reduced in weight. It must be remembered, however, that the total weight

of the engine involves many parts in which the limiting strength is determined by inertia considerations, which are independent of gas pressures; also many parts must be dimensioned with a view to casting considerations and certain minimum thicknesses must be allowed, so that the reduction of weight obtainable by so-called light and over-sized design is not so great as might be supposed.

Further interesting features of the Napier Lion are the use of roller bearings for the crankshaft to reduce bearing loading difficulties as in the R.A.F. engines, and Hispano camshafts operating directly on the valve stems.

Type 4, the Cross type, did not come into service during the war, though engines were constructed in England, by Messrs. Leyland Motors and the Aircraft Manufacturing Company; in France by Clerget and Anzani; in America by the Trebert Company.

The rotary air-cooled engine, Type 5, was very largely used, and up to powers of about 230 B.H.P. has been found to be quite satisfactory. The particulars of the chief engines of the type are shown in Table 12.

TABLE 12.

ROTARY ENGINES.

Make of Engine.		Gnome	Monosoupape Gnome		Le Rhone	Clerget.	B.R.1	B.R 2.
Rated Horse-Power	H.P.	80	100	150	110	130	150	200
Number of Cylinders	No.	7	9	9	9	9	9	9
Bore of Cylinders	Ins.	4.88	4.33	4.55	4.41	4.72	4.72	5.51
Stroke of Pistons	Ins.	5.51	5.91	6.69	6.69	6.30	6.69	7.08
Total Swept Volume of Engine ..	Cu.In.	748	783	987	920	994	1055	1522
Compression Ratio	R.	4.30	4.90	5.42	4.82	4.56	5.09	5.30
Brake Horse-Power	B.H.P.	68.5	104	152	110	127	166	238
Revolutions per Minute	R.P.M.	1150	1200	1250	1200	1250	1250	1300
Piston Speed in Feet per Minute	Ft./Mn.	1095	1180	1392	1340	1312	1394	1536
Brake Mean Pressure	B.M.P.	83.0	87.7	97.6	78.9	81.0	94.0	95.2
Fuel per B.H.P. Hour	Lbs.	0.787	0.690	0.634	0.560	0.60	0.59	0.63
Oil per B.H.P. Hour	Lbs.	0.190	0.180	0.112	0.091	0.123	0.109	0.074
Brake Thermal Efficiency. (18600 B.T.U. per lb.)	%	17.4	19.8	21.6	24.4	22.8	23.2	21.7
Weight of Engine Complete ..	Lbs.	224	288	313	336	400	405	498
Weight per B.H.P.	Lbs.	3.27	2.77	2.06	3.06	3.15	2.60	2.10

The early Gnome type was followed by the 100-horsepower Monosoupape Gnome, Fig. 16. The distinctive feature of this engine is the induction arrangement by which rich mixture is introduced to the cylinder from the crankcase through ports uncovered by the piston, the charge being made up by air taken in through the exhaust valve in the cylinder head. This valve is operated by a push rod by means of nine cams driving from the rotating crankcase by an epicyclic gear. A noteworthy feature of this construction is the low gas velocity obtained by the use of the single valve. Subsequently an engine of 150 horsepower of similar design was built and gave good results.

The Le Rhone engine, which was also largely used, has been already described, and except as regards power has undergone little modification.

The Clerget engines are noteworthy in respect of the valve operating gear employed, which has been adopted in the later BR1 and BR2 engines. The cylinders are machined from solid steel billets, as in the Gnome engines.

The BR1 engine, designed in Great Britain, is very similar to the Clerget, the only substantial novelty in its design being the construction of the cylinders, with cast aluminium barrels shrunk on to cast iron, and later steel liners. The cylinder heads are machined in steel with the induction pockets in one piece, and are held down by bolts inserted from the inside of the crankcase. The design of the cooling surface leaves something to be desired in comparison with more recent air-cooled cylinders. This engine, however, showed good reliability in service.

The BR2 engine, developed from the BR1 to meet demands for increased power, differed from the earlier BR1 in the use of steel instead of cast iron for the liners; in other respects the design is similar.

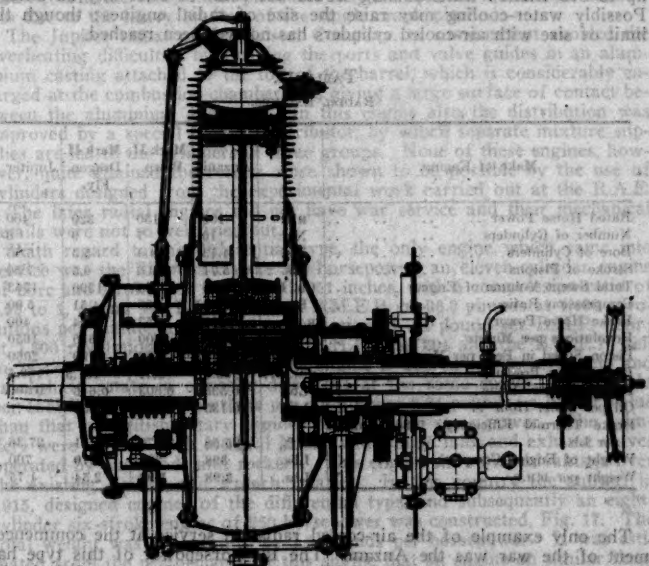

FIG. 16.

With regard to engines actually in service during the war, it may be said that development of the rotary has merely taken the line of eliminating obvious defects, and improving the cylinder construction to a certain extent by suiting the design to the use of materials of higher heat conductivity. Improvement in design in this respect was not, however, carried to the extent which has been shown to be possible by the experiments carried out at the R.A.E. Farnborough. The rotary type was not accepted as a final type, and thus escaped to some extent the close attention given to other designs. The gyroscopic effects and heavy windage losses are against it.

It may be said that the rotary did not develop in production many points of novelty. The peculiar cycle of the single valve Gnome is one to which attention may be drawn. This principle may possibly be further developed.

The radial engines, Table 13, offer advantages as regards weight for power. It has been shown that such engines, even on our present knowledge, can be built of a lower weight per B.H.P. than any of the other types, and although their reliability has not as yet come up to the standard reached by the water-cooled vertical and Vee types, their advantage in weight leaves open possibilities of their attaining considerable commercial use, with a weight per horsepower lower than that obtainable with other types, which have already been developed to a more advanced stage. Water-cooling may be employed in radial engines, as in the Canton Unné, which was extensively used early in the war, but the radial type offers such great advantages as an air-cooled engine that for engines of powers up to 500 B.H.P. water-cooling would not seem to offer any advantage. Possibly water-cooling may raise the size of radial engines; though the limit of size with air-cooled cylinders has not yet been reached.

TABLE 13.

RADIAL ENGINES.

Make of Engine.		Anzani.	Mark II. Wasp.	Mark II. Dragon Fly.	Jupiter.
Rated Horse Power	B.H.P.	100	150	320	400
Number of Cylinders	No.	10	7	9	9
Bore of Cylinders	Ins.	4.13	4.5	5.5	5.75
Stroke of Pistons	Ins.	5.71	6.0	6.5	7.50
Total Swept Volume of Engine	Cu. Ins.	766	988	1390	1753
Compression Ratio	R.	4.48	4.00	4.51	5.00
Brake Horse Power	B.H.P.	100	153	294	400
Revolutions per Minute	R.P.M.	1200	1600	1650	1650
Piston Speed in Feet per Minute	Ft./Min.	1140	1500	1790	2060
Brake Mean Pressure	B.M.E.P.	86.0	99.1	101.5	109.5
Fuel per B.H.P. Hour	Lbs.	0.535	0.603	0.645	0.500
Oil per B.H.P. Hour	Lbs.	0.135	0.030	0.030	
Brake Thermal Efficiency, (18600 B.T.U. per Lb.)	%	25.58	22.70	21.21	27.36
Weight of Engine Complete	Lbs.	398	294	670	700
Weight per B.H.P.	Lbs.	3.98	1.92	2.54	1.75

The only example of the air-cooled radial in service at the commencement of the war was the Anzani. The 125 horsepower of this type has ten cylinders arranged in groups of five about a two-throw crankshaft, the bore and stroke being 4.53 inches and 5.93 inches, respectively, with a brake mean pressure of 86.2 pounds at 1,200 revolutions per minute. The fuel consumption was not excessive, being only .57 pound per B.H.P. hour; a very creditable performance with the knowledge of air-cooling available at that date. It will be noticed that this two-throw type of radial has the advantage that the balance weights necessary in the single row radial are not required. The weight was, however, somewhat high, amounting to 3.7 pounds per horsepower, due to the low B.M.E.P. which was 86.2; with a B.M.E.P. of 110 pounds per square inch, the weight per horsepower would become 2.9 pounds.

Very early in the war an air-cooled radial, the Smith Static, was used to some extent. The engine had 10 cylinders, 4.53-inch bore and 6.10-inch stroke. These were all placed in one plane; slanting connecting rods were used working on a two-throw crankshaft. A special fan arrangement was used for cooling. The compression ratio was 4.5 to 1 and the normal

B.H.P. 150; this power being developed at 1,300 revolutions per minute. The fuel consumption was 0.53 pound per B.H.P. hour and the oil consumption 0.062 pound per B.H.P. hour. The B.M.E.P. was 92 pounds per square inch.

Later engines of the radial type, the Wasp and the Dragonfly, which were not brought into full service use, show considerable advance. In these engines, one-piece solid-ended steel cylinders are employed with two exhaust valves and one inlet valve per cylinder. The heads are flat and of the same diameter as the barrels, with separate cast iron and aluminium castings for the exhaust and inlet ports and valve guides. Cylinders designed at the R.A.E. and fitted to these engines have given much improved results. The increased mean pressures may increase the mechanical difficulties due to the heavy loading of the single crank pin.

The Jupiter radial of the Bristol Company, to some extent avoided the overheating difficulties by forming the ports and valve guides in an aluminium casting attached to the top of the barrel, which is considerably enlarged at the combustion chamber end, giving a large surface of contact between the aluminium and steel. In this engine also the distribution was improved by a special type of distributor, by which separate mixture supplies are led to the cylinders in three groups. None of these engines, however, quite attained the results since shown to be possible by the use of cylinders designed from the experimental work carried out at the R.A.E. The later radial engines did not have war service and their mechanical details were not so well tried out.

With regard to the differential type, the only engine which came into service was the Siemens-Halske 200-horsepower, an eleven-cylinder engine of bore and stroke 4.88 inches and 5.51 inches, with a compression ratio of 4.95 to 1, giving 193 B.H.P., with a B.M.E.P. of 96.2 pounds at 1,400 revolutions per minute, and a fuel consumption of .64 pound per B.H.P. hour; at 1,800 revolutions per minute the B.H.P. was 204 and the B.M.E.P. fell to 79 pounds per square inch. The engine is geared so that cylinders and crankshaft rotate in opposite directions at the same speed. The weight per B.H.P. at 1,600 revolutions per minute was 2.24 pounds; a higher value than that of British rotary engines of about the same power. The cylinders were of steel with integral heads, with single inlet and exhaust valves operated by push rods and rockers. This engine shows no advantage over the existing British and French rotaries. In this country the author, in 1915, designed engines of the differential type, and subsequently an eight-cylinder six-stroke engine of 250 horsepower was constructed, Fig. 17. The total weight was 400 pounds, or 1.6 pounds per horsepower. It was estimated from the performance of this engine that an engine on similar lines, working on the four-stroke cycle could be constructed at the extremely low weight of 1.11 pounds per B.H.P. Subsequently, Dr. Blackwood Murray, using the designs and results of previous experiments, designed an engine to give 600 horsepower with a piston speed of 2,500 feet per minute, and weight of 610 pounds, or just over 1 pound per B.H.P. This was not, however, proceeded with.

It will now be of interest, before dealing with matters which may be said to be still experimental, to tabulate the results obtained in the different types by engines which had reached the stage at which production in quantity was considered expedient. The results given in Table 14 are examples selected as being approximately the maximum obtained in ordinary practice in production engines at the end of the war.

Many engines of various experimental types have been built during the war, but have not yet reached the stage of development at which production is justified.

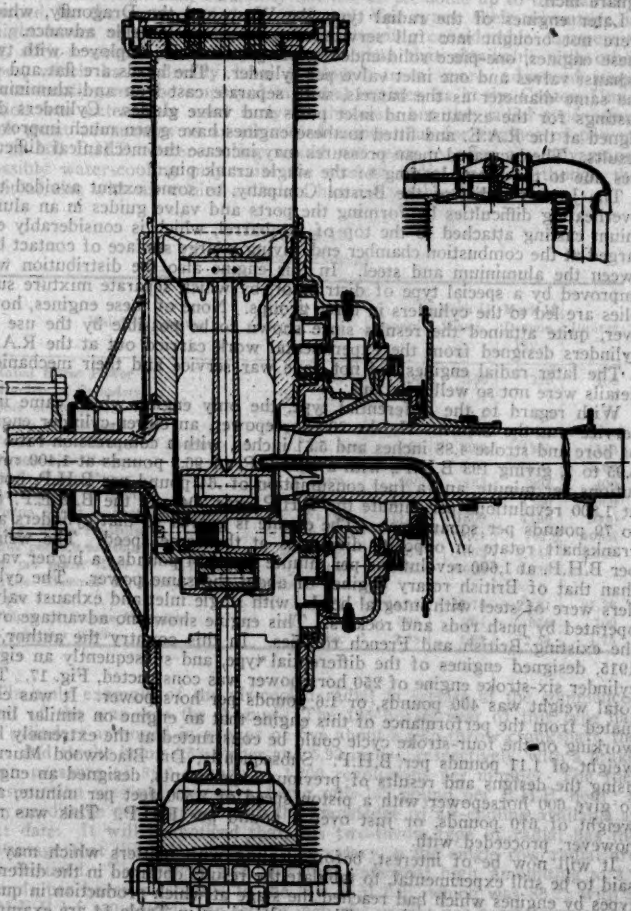

FIG. 17.

One important class of these has been designed with the object of producing engines capable of approaching their normal ground level output when working at the lower atmospheric densities existing at high altitudes.

Four main methods of effecting this have been proposed:—

(1) Supercharging, *i.e.*, admitting to the cylinder an additional charge of air or exhaust products or both at the end of the suction stroke at a pressure of a few pounds above the existing atmospheric pressure.

(2) Forced induction, compressing air taken in at the existing atmospheric density before leading it to the cylinder so that the inlet pressure at high altitude approximates to atmospheric density at ground level.

(3) Design for low atmospheric pressure with restricted output at ground level.

(4) Increasing the compression ratio at altitude by varying the stroke or other means.

TABLE 14.

SOME COMPARATIVE RESULTS.

Type of Engine.		Vertical.	Vee.		Vee.		Arrow.	Rotary	Rotary	Radial
Cooling System ..		Water	Water		Air.		Water	Air	Air	Air
Number of Cylinders	No.	6	8	12	8	12	12	9	9	9
Bore of Cylinders ..	Ins.	5.71	5.51	4.50	3.94	3.94	5.50	4.51	5.31	5.75
Stroke of Pistons ..	Ins.	7.48	5.91	6.50	5.51	5.51	5.125	6.00	7.08	7.50
Brake Horse Power	B.H.P.	240	300	350	90	196	450	152	238	400
Revolutions per Min.	R.P.M.	1400	1800	1800	1600	1800	2000	1250	1300	1650
Brake Mean Pressure	lbs sq"	118.1	117	124	83	107	122	97.6	95.3	100.5
Piston Speed in Feet per Minute ..	Ft.Mn.	1746	1770	1950	1470	1654	1708	1392	1536	2060
Fuel per B.H.P. Hour	Lbs.	0.500	0.587	0.50	0.700	0.532	0.495	0.634	0.630	0.500
Oil per B.H.P. Hour ..	Lbs.	0.002	0.040	0.028	0.056	0.054	0.022	0.112	0.074	—
Weight per B.H.P. complete with Cooling System ..	Lbs.	3.25	2.27	3.3	5.20	3.42	2.52	2.06	2.10	1.75

The first method is a development of the air and exhaust gases supercompression gas engines proposed by Sir Dugald Clerk in order to obtain high mean pressures with maximum pressures and temperatures lower than those reached in normal four-stroke engines when working on gases of high hydrogen content, particularly in large cylinders. In Clerk's exhaust supercompression engine the piston was arranged to overrun ports round the cylinder at the outer end of the stroke. The exhaust gas passed from the cylinder through these ports to a water jacketed annular space and through a long cooled pipe to a reservoir, a sufficient quantity of exhaust gas being allowed to pass to keep the pressure in this reservoir above that of the atmosphere. A piston valve was arranged to close the inlet to the reservoir before the main exhaust valve opened. On the next stroke, immediately after the charge inlet valve was closed the piston valve was opened allowing some of the cooled exhaust gas from the reservoir to flow by another pipe to the cylinder raising the pressure to about 3 pounds above atmosphere. Clerk also suggested that temperature in large gas engine cylinders might be kept down by scavenging with cooled exhaust gas.

During the war proposals were made to apply supercharging to a B.H.P. engine and also to a Brotherhood engine but did not lead to further construction. Subsequently an engine was designed by Mr. H. Ricardo and built in the form of a 300-horsepower 12-cylinder water-cooled Vee

engine with the banks of cylinders inclined at 30 degrees. A crosshead piston is employed and alternately draws in and compresses air in the closed space below it; the piston overruns ports in the cylinder as in the Clerk engine and a portion of the exhaust products pass through the cooled passage to the space below the piston into which the air has been drawn during the previous compression stroke. No valve other than the ports overrun by the piston is employed to control the flow of the exhaust products from and to the cylinder and air is drawn in through an automatic valve during each instroke of the piston. The cooled exhaust gases form only a small proportion of the super-charge which consists mainly of air, and the hot gases which would normally remain in the cylinder at the end of the exhaust stroke are displaced by cooled exhaust gas and a considerable proportion of air. A single cylinder unit of 110 mm. bore and 140 mm. stroke with a compression ratio of 5 to 1 running at 1,500 revolutions per minute developed 16.6 B.H.P. without supercharge and 22.4 B.H.P. with supercharge and correspondingly increased fuel supply at the same speed.

With regard to the second method, forced induction, a considerable amount of work has been done at the R.A.E., Farnborough, and also in Germany and the United States.

Good results have been obtained with a blower running at a high speed mounted on the shaft of an exhaust driven turbine. This was first proposed by Prof. Rateau in France, and promising results have been obtained with experimental units at the R.A.E. and also in France and the United States. A blower of this type applied to a Liberty engine has been tested at the R.A.E. and has shown that ground level power can be maintained to heights of 20,000 feet or more.

This method will probably be found the most suitable for very high altitudes.

As regards the third method, engines have been designed with a specially high compression ratio for altitude and some of these have come into service, for example, the Napier Lion in this country and many airship engines, and also the Bayern engine in Germany. So far as the author is aware, the compression has not been carried to the high value which would be possible with satisfactory means for limiting output at ground level, though for moderate altitudes it would appear to provide the most satisfactory solution of the problem. With regard to the fourth method, variable compression, so far as the author is aware, there were few, if any, aero engines in operation in which variable compression was employed.

Many experimental types of engines were proposed during the war; but for the most part they would have required prolonged research and experiment before they could reach the stage of development at which production in quantity could be justified. It may be of interest to mention a few of these proposals.

Although no aero engine working on the two-stroke cycle has reached the production stage, a number of different types have been proposed. Engines working on the two-stroke cycle were built by the New Engine Company in 1912. These engines were produced in two sizes, namely 50 horsepower and 100 horsepower. The 50 horsepower was a Vee type engine with 4 cylinders 3 11/16-inch bore and 4½-inch stroke, developing 50 B.H.P. at 1,250 revolutions per minute. The 100-horsepower engine had 6 vertical cylinders of 4-inch bore and 5½-inch stroke. The weight of the engine with silencers bolted to the cylinders was 330 pounds. It developed 100 B.H.P. at 1,250 revolutions per minute. The fuel and oil consumption of these engines was stated to be approximately 0.58 pound and 0.026 pound per B.H.P. hour, respectively. The mechanism provided for pumping in the fresh gas and driving out the exhaust was a special form of Roots

Blower; it was divided so that one part dealt with pure air and the other with gas. A rotary valve, interposed between the blower and the cylinder controlled the inlet and air only was admitted until all the exhaust products were expelled.

Another engine which may be mentioned is the Fredickson two-stroke rotary which was built in the United States with 3, 5, and 10 cylinders. In this engine the lower part of the cylinder is partitioned off to form a space in which the charge taken in from the crankcase is compressed. An oscillating slide valve controls the passage of the mixture from the crankcase to the compression chamber and from this to the cylinder.

Prof. Junker, in Germany, has constructed a high speed two-stroke cycle engine with two pistons moving in opposite directions in each cylinder and controlling ports in the cylinder at the outer end of their stroke. Pure air scavenging and direct fuel injection are stated to be employed, a high pressure fuel pump regulating the mixture. A number of cylinders are placed parallel to each other with two parallel crankshafts at the ends of the cylinders. The propeller shaft is driven by a spur wheel gearing with spur wheels on the crankshafts.

The cooling of the pistons is effected by oil enclosed in cavities within them, so that the oil in violent motion carries the heat from the head of the piston to its sides and so to the cylinder walls.

Hirth is also stated to have tested a two-stroke cycle engine in which the exhaust gases are drawn off through a hollow steel propeller which at the same time serves to draw in the fresh charge; separate exhaust and inlet ports are successively uncovered by the piston.

The engine is intended to run at 2,400 revolutions per minute, giving 300-400 horsepower with a weight per horsepower of 1.65 to 1.76 pounds.

Another interesting experimental type is the swash-plate engine, one of which, known as the Perfetti Dolara, has been constructed in Italy in the form of a nine-cylinder air-cooled engine of 130 mm. bore by 176 mm. stroke. It is stated to give 300 B.H.P. at 1,500 revolutions per minute, with a weight of 2.05 pounds per horsepower.

Another of this type is the Motor La Meuse, a six-cylinder water-cooled stationary engine, 120 mm. bore by 140 mm. stroke, with cylinders parallel to the shaft axis and developing 125 horsepower.

Oscillating cylinders have been employed in the Seja engine.

Lecture III.—Delivered January 31st, 1921.

There were several very interesting types of engines constructed during the latter days of the war or shortly after, some of very high power.

Of the 12-cylinder Vee type, there are the Galloway "Atlantic," which is substantially two "B.H.P." engines arranged as a Vee; the Siddeley Tiger, which has cylinders 160 m/m bore and 180 m/m stroke, giving 500 B.H.P. at 1,500 revolutions per minute; the Rolls Royce "Condor," with cylinders having a bore and stroke of 5.5 inches and 7.5 inches, respectively; the B.H.P. developed is up to 600 at 1,750 revolutions per minute. Another large engine is the Sunbeam Sikh, which is stated to develop 850 horsepower at 1,400 revolutions per minute; the cylinder bore is 180 m/m and the stroke 210 m/m.

In France, Peugeot has produced an engine with a bore of 160 m/m and stroke of 170 m/m, which develops 600 horsepower at 1,600 revolutions per minute.

With regard to 8-cylinder Vee engines, the Hispano-Suiza now develops well over 300 B.H.P., and is the leading engine of this type.

The three-row or arrow type is now built by Lorraine Dietrich with 24 cylinders, having a bore of 120 m/m and stroke of 200 m/m. This engine develops 1,000 horsepower.

The Cross type has been built by Peugeot in the form of a 16-cylinder water-cooled engine of 130 m/m bore by 170 m/m stroke.

A very interesting engine, completed just before the termination of the war, was the Cross type designed by J. G. P. Thomas, of Messrs. Leyland Motors. (Figure 18.) This engine had 8 cylinders; developed 420 horse-

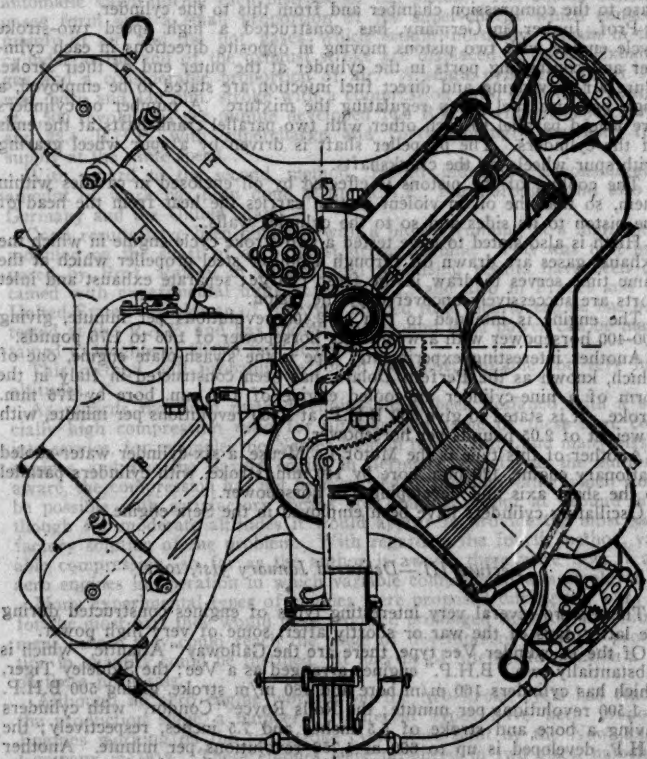

FIG. 18.

power at 2,500 revolutions per minute, and its dry weight, including dynamotor for starting purposes, was 550 pounds, or 1.31 pounds per horsepower.

Of the Radial type reference may be made to the Armstrong-Siddeley, Anzani and F.I.A.T. engines.

Table 15.

Comparative Table of Types.

Type of Engine.	Name of Engine.	Performance at Normal Piston Speed and Normal Brake Mean Pressure.					Performance with Normal Piston Speed and 130 lbs./sq. in. B.M.E.P.			Performance with Normal B.M.E.P. and Piston Speed of 2,000 Ft./Min.			Performance with Piston Speed of 2,000 Ft./Min. and B.M.E.P. of 130 lbs./sq. in.		
		Weight per B.H.P. in Lbs.	R.P.M.	Piston Speed Feet/ Min.	B.M.E.P. Lbs. per sq. ins.	B.H.P.	Weight Lbs./ B.H.P.	R.P.M.	B.H.P.	Weight Lbs./ B.H.P.	R.P.M.	B.H.P.	Weight Lbs./ B.H.P.	R.P.M.	B.H.P.
Vertical Water Cooled	"Puma"	3.25	1400	1746	118.1	240	2.96	1400	264	2.84	1605	275	2.50	1605	312
Vee Water Cooled	300 H.P. Hispano Suiza	2.65	1800	1770	110.8	300	2.38	1800	334	2.34	2160	339	1.99	2160	400
Vee Water Cooled	Rolls Royce Eagle	3.37	1800	1950	124.0	350	3.21	1800	367	3.28	1847	359	3.13	1847	376
Vee Water Cooled	Liberty	2.69	1650	1925	118.0	405	2.44	1650	446	2.59	1715	430	2.34	1715	464
Arrow Water Cooled	Napier "Lion"	2.57	2000	1740	122.0	450	2.40	2000	480	2.19	2340	526	2.06	2340	561
Rotary Air Cooled	B.R.2.	2.10	1300	1530	95.2	238	1.53	1300	325	1.60	1600	310	1.17	1600	425
Radial Air Cooled	Cosmos "Jupiter"	1.75	1650	2060	109.5	400	1.47	1650	475	1.80	1600	388	1.52	1600	460

The Armstrong-Siddeley is built in two sizes: 150 horsepower and 300 horsepower. These engines have a bore and stroke of 127 m/m; they are both air-cooled and each develops its stated power at 1,500 revolutions per minute. A water-cooled radial with twenty cylinders and capable of developing 600 horsepower, has been produced by Anzani. In Italy the F.I.A.T. have constructed a water-cooled radial engine with 9 cylinders, 130 m/m by 150 m/m, which develops 300 horsepower at 1,800 revolutions per minute.

Other types and modifications of the main types have been proposed, and it will be obvious that many cylinder arrangements are possible, and no doubt, when the commercial prospects of aviation become more easy, we shall see a great renewal of activity in the production of new models.

In dealing with the future development of the aero motor, I think we may consider it with relation to a comparison of types of existing engines, as given in Table 15.

We may discuss the progress that may be made under three headings:

(1) Thermodynamical.
(2) Mechanical.
(3) Metallurgical.

Taking thermodynamic considerations first, the present aero engines work almost universally on the four-stroke constant volume cycle, with equal compression and expansion, and any departures which have been made from this cycle may be regarded as purely experimental.

The best aero engines at the present day obtain thermal efficiencies of over 30 per cent; high compression land engines have reached 36 per cent. It is obvious that within limits imposed by mechanical considerations, the thermal efficiency is increased by raising the compression ratio, but this is already in most cases as high as the fuel employed will permit.

In the constant volume cycle, the methods which offer a prospect of improving the thermal efficiency are:—

(1) Extending the expansion ratio keeping the maximum allowable compression ratio. This can be effected by:—
  (*a*) Mechanical devices for varying the stroke.
  (*b*) Compounding.
  (*c*) Allowing greater expansion than compression in the cylinder.
(2) Raising the compression ratio, made possible by the use of a fuel less liable to detonate.
(3) Supercharging.
(4) Stratified working by which losses due to cooling and variable specific heat and dissociation are reduced.
(5) Regeneration, *i.e.*, utilizing heat lost to the jacket and exhaust.
(6) Constant pressure and dual cycles, etc.
(7) Direct injection.

Taking these methods in order, the first, (1*a*), can be carried out by varying the piston travel during successive strokes by the use of suitable mechanical devices. An early example of this is Atkinson's link motion (see Fig. 6), and although it is stated that high mechanical efficiency was obtained in the early low speed gas engines, in which it was originally employed, the mechanism as used is hardly suited to aero engine practice, though it has the remarkable property of giving four strokes of different length per revolution for a four-stroke cycle, thus allowing high piston speed and low propeller speed.

Another device is an eccentric mounted on the crank-pin and rotated in suitable phase; this method can be readily applied to a radial engine. A variable stroke was obtained in one of the Clerget engines, and in the Zeitlin.

With regard to (1*b*), compounding, many attempts have been made to build a compound internal combustion engine, but so far with little success, even for heavy stationary engines; the difficulty has always been the loss of heat in transference of the working fluid to the low pressure cylinder. For aero work the large size of engine required to utilize the low pressures would make this method quite unsuitable. Better results are likely to be obtained by method (1*c*), allowing extended expansion in the same cylinder. This may be effected by late closing of the inlet valve during the compression stroke; this would, of course, involve the use of a lower compression ratio, unless the bearings can be raised by means of an eccentric mounting as a compensation.

Mr. Ricardo has suggested that this method should be employed to enable an engine, working with a very high compression at altitude, to be run at ground level at a lower compression without undue loss of efficiency.

The economy can be safely increased by raising the compression ratio above present ratios if special fuels are employed containing a higher proportion of toluene or other aromatic hydrocarbons. By using fuels containing suitable proportions of these compounds, the possible compression ratio at ground level may be raised from 4.85, the highest permissible with a light paraffin petrol, to 7.05, giving an increase in indicated thermal efficiency of from .311 to .373. Above this ratio we come to conditions in which, up to the present, only direct injection of fuel has overcome the pre-ignition and other difficulties. This is dealt with later. The addition of cooled exhaust products is another method which enables the compression ratio to be raised, or more difficult gases or fuels to be worked with. It was used by Clerk in 1902, added separately, and in Germany it was mixed with the gas previous to admission. The author was concerned with its use for admixture with coke oven gas in these early days. This gas has a large percentage of hydrogen, and is extremely prone to pre-ignition, etc.; it is quite readily used when mixed with cooled exhaust products at normal compressions. In considering the use of hydrogen for airship engines made available by the reduction of weight owing to the amount of fuel used, the use of cooled exhaust products as a diluent should not be lost sight of. Mr. Ricardo has proposed an ingenious form of altitude control by its use.

Supercharging, although it has hitherto been very little used, is undoubtedly likely to be employed again both for economy, increase of power, and for altitude correction; for high altitude work the alternative method of forced induction will also undoubtedly come into use. In both, the mass of working fluid in the cylinder is increased. Pumps or rotary blowers may be employed or the required compressed air or cooled exhaust products or mixture of these for the extra charge may be obtained by utilizing the exhaust pressure in the engine cylinders.

In a six-cylinder-in-line engine this may be effected by using the exhaust pressure of one cylinder to blow air and cooled exhaust products into another cylinder at the phase of its cycle at which the extra charge is required. The thermodynamic economy in a supercharging engine is due to the lowering of the temperature of the working fluid while the mean pressure is kept high, so that the losses due to cooling and to the change of specific heat and dissociation are minimized.

STRATIFIED WORKING.

The idea of stratified working is a very old one and was first raised by Otto and his supporters in an attempt to show that in the original Otto engines, also early two-cycle engines, there existed at the moment of ignition a layer of inert gas in the neighborhood of the piston. In the ordinary high speed four stroke or two-stroke petrol engine there can be no

doubt that owing to turbulence this is extremely difficult to obtain, for the cylinder at the moment of ignition contains a substantially homogeneous mixture. The use of a specially shaped cylinder head and proper control of the admission charge is one method of overcoming this and is used in one form by Ricardo. With delayed gas admission there is always a sufficiently rich mixture to ignite in the neighborhood of the sparking plugs, while in the neighborhood of the piston the mixture is extremely weak or air alone. This makes possible the complete combustion of relatively weak mixtures, whereby the temperature in the engine cylinder can be very much reduced, so that, whereas in the ordinary cycle with fuel sufficient to burn the whole of the oxygen in the charge, the maximum temperature rises to approximately 2,500 degrees C., by means of stratification, the temperature, and with it the cooling and specific heat and dissociation losses, can be so much reduced that higher efficiencies can be obtained. In an engine with a compression ratio of 5 to 1 working with stratified charge at one-third full load a consumption of only 0.36 pint of benzol per indicated horsepower hour has been obtained. It will be seen that this method of working involves the use of large cylinders in proportion to the power obtained when running with maximum economy. Inversely it allows the power of an engine to be reduced with maximum economical working, a condition called for when the aeroplane is at flying elevation after leaving the ground. In another form used with two-cycle engines, extraordinarily good results were obtained by the author in 1913; these are given in Table 16. The engine was a large one and the speed slow; this allowed the stratification to to be carried out with more than usual accuracy.

TABLE 16.

STRATIFIED CHARGING.

RESULTS OF EXPERIMENTS MADE ON A LARGE GAS ENGINE.

Two-Stroke-Cycle-Light Load-Slow Speeds.

Date.	Average B.T.U's per I.H.P. hour	Indicated Horse Power.	Mean Effective Pressure.	Revs. Per Minute.	Indicated Thermal Efficiency.
3-7-13	5620	83.5	30.4	87.0	45.2%
3-7-13	5800	83.2	31.4	87.0	43.8%
3-7-13	6020	81.7	29.9	87.0	42.2%
3-7-13	5930	87.0	31.0	89.0	42.8%
Average	5843	83.8	30.7	87.5	43.5%

Approximate Contents of Working Cylinder.
Volume of Cylinder = 8.49 cubic feet.
Volume of Mixture = 3.44 cubic feet.
Volume of Exhaust Products = 5.05 cubic feet.

REGENERATION.

The great loss of heat to the jackets and by the exhaust has directed the attention of many designers to methods of conserving it. In stationary engines it has been proposed to employ the jacket water and exhaust heat to raise steam. In the Still engine the cylinder heat is utilized by using

the generated steam on the other side of the internal combustion piston. There is no doubt that increased economy can be obtained in this way, but much experiment is required before this method can be adapted to aero engine work and the additional weight required will probably limit the use of the method to airships alone. Methods of using air as the heat transferring agency, to be added to the compressed cylinder air at the top of the cycle, have not been developed.

It is of interest to note that in the operation of exhaust driven turbine blowers for forced induction in high altitude engines, regeneration is to a certain extent employed owing to the conversion of a portion of the exhaust heat into kinetic energy in the turbine nozzles.

The constant pressure cycle has so far been very little employed, if at all, in aero engines. Owing to the higher compression which can be employed when air alone is compressed higher thermodynamic efficiencies can be obtained, but at the expense of considerably increased weight due to the much higher pressure. This cycle will probably find its most useful application in airship work where economy of consumption is of great importance and weight is of less account. In any case the type of engine most likely to be employed works probably more on the constant volume than on the constant pressure cycle. This type does not employ air to inject the fuel as in the Diesel.

Direct fuel injection is employed in all high compression engines, and as it can be used for other types as well, it has become of very great importance as a system for thorough investigation; and in addition to its advantages as regards performance, it materially reduces the risk of fire.

In the first type, in which the engine is of substantially the same strength and type as those in common use, the fuel is usually injected on the suction stroke or early in the compression stroke. With suction stroke injection remarkably high mean pressures have been obtained (over 130 pounds per square inch); the economy so far is probably not up to that obtained with the usual carburetor; later injection on the compression stroke tends to reduce the mean pressure as more penetration is involved, while the turbulence is decreasing. Finality has, however, by no means been reached, and considerably more experimenting is required; the results so far obtained are of good augury. A further advantage of the system is that a common inlet and exhaust valve can be employed, allowing a large valve area and good volumetric efficiency for piston speeds up to 3,000 feet per minute and more; also facilitating the cooling of the valves and cylinder head.

Direct injection was due to Akroyd-Stuart, and, therefore, he may be said to be the originator of all high compression engines, including the Diesel.

The Blackstone engine uses two jets and works on the so-called "Dual" cycle.

In the higher compression types, including the Diesel, the compression may be up to 500 pounds per square inch, and even over, with consequent very much increased engine weight. Unless this can be considerably reduced, the only field seems to be for airships. In the Vickers type, as used on submarines, where the compression is 380 to 400 pounds per square inch, the remarkably low fuel consumption of .385 pound per B.H.P. hour has been obtained. What is also very valuable for long distance work, crude oil can be used with considerable saving in fuel cost. Although no airship engines are at work on this cycle in this country as yet, it is reported that they have been built in Germany.

Mechanical development will take place in raising output relatively and reducing weight per B.H.P. by overcoming the difficulties which impose limits to the speeds at which engines can now be run, which as previously mentioned, are reached in practically all types due to the bearing loadings and valve gear, etc.

Further, and most importantly by increasing the reliability by the gradual removal of all risks from over stressed parts or those still unknown.

With regard to the former, the difficulties of high speed increase with the thermodynamic and volumetric improvement, since the higher mean pressures involve increased strength and weight of connecting rods, big-ends, etc., with correspondingly increased inertia forces and bearing loadings. Increase of compression ratio particularly approaches a limit at which any increase in thermodynamic efficiency involves such increased weights of moving parts that the advantage gained is more than balanced by the mechanical disadvantages involved. We may, with advantage, turn again to Table 15. This table has been calculated for a common mean pressure of 130 pounds per square inch and piston speed of 2,000 feet per minute. The increased speed difficulties are the greater, and it is important to compare their effects on different types. In the main, they refer to big-end bearings, main bearings, and valve gear. The difficulties of heavy bearing loading are most pronounced at the crankpin and become specially important in types with multiple big-ends, such as the radial; but highest at the crankshaft in vertical, Vee and other types. The use of roller bearings, as in the R.A.F., Napier Lion, etc., has largely overcome these difficulties, though speeds and loads may possibly be further increased to a limited extent by the use of higher pressures for lubrication.

It is of interest to consider what are the actual limiting factors in engine speed.

(1) Inertia forces and bearing loadings.
(2) Provision of adequate valve area.
(3) Acceleration forces in the valve gear.
(4) Inertia of the connecting rod.
(5) Rate of heat flow as affecting jacket walls, piston, valves and sparking plugs.
(6) Ignition gear.

It is not proposed to discuss all these factors in relation to the limiting speed, the primary object being to draw attention to the limitation imposed by the bearing loadings; it is, however, of interest to note that with the exception of item (6) and, to a certain extent, item (2), increased speed seems to point to the necessity of using small cylinders. This is especially the case as regards the inertia forces since the masses concerned are invariably machined all over and their weight varies therefore as the cube of their dimensions.

To direct attention to the importance of the big-end and its limiting effect on speed, the following approximate figures have been kindly worked out at the request of the lecturer, by Mr. H. A. Hetherington. They reveal the state of affairs very clearly; it will be seen that the simple six can be run very much faster than the single-plane radial, a point hardly sufficiently realized when comparing types. Further, if other considerations did not in turn become limiting features, it is indicated that there is less difference between weights of various types than is usually supposed.

In examining this important limitation let us assume four engines, each of 1,200 cubic inches capacity, stroke/bore ratio = 1.17:1 and running at a piston speed (v) of 2,000 feet per minute.

The dimensions of the engines will be assumed to be as follows:—

	Bore, inches.	Stroke, inches.
(*a*) Six-cylinder vertical	6.00	7.00
(*b*) Twelve-cylinder Vee	4.78	5.60
(*c*) Twelve-cylinder Fan	4.78	5.60
(*d*) Nine-cylinder Radial	5.25	6.14

The equivalent revolution speeds corresponding to a piston speed (v) of 2,000 feet/minute will be given by

$$n = 12000/s$$

which gives us

(a) = 1,715 revolutions per minute.
(b) = 2,140 revolutions per minute.
(c) = 2,140 revolutions per minute.
(d) = 1,960 revolutions per minute.

The rotating weight acting on the big-end bearing will be assumed to be equal to

$$\text{Wt.} = 0.18 \times A \times \text{Constant},$$

A being the area of one piston in square inches, and the constant varying according to the number of pistons coupled to one crankpin.

With the above assumptions we arrive at the following values for the rotating weights (w) acting on the big-end bearing.

(a) $0.18 \times 28.4 \times 1 = 5.1$ pounds.
(b) $0.18 \times 18.0 \times 1.8 = 5.82$ pounds.
(c) $0.18 \times 18.0 \times 2.3 = 7.45$ pounds.
(d) $0.18 \times 21.7 \times 4.2 = 16.4$ pounds.

The centrifugal pressure (cp) due to these rotating masses is given by the formula:—

$$cp = 0.000142 \times w \times n^2 \times s$$

which gives us the following values for the four types of engines under consideration:—

(a) 1,490 pounds.
(b) 2,120 pounds.
(c) 2,720 pounds.
(d) 5,480 pounds.

In order to get some idea of the actual loading on the big-end bearing, due to this pressure, we must further assume some dimensions for the crankpin which agree reasonably well with average modern practice. The dimensions which have been assumed are tabulated below for the four types of engines.

	Ins. (a)	(b)	(c)	Ins. (d)
Bore of cylinder	6.0	4.78	4.78	5.25
Diam. of crankpin	2.52	2.15	2.20	2.62
Ratio $\frac{\text{Diam. crankpin}}{\text{Bore of cylinder}}$	0.42	0.45	0.46	0.50
Effective length of big-end bearing (inches)	2.9	2.3	2.5	3.1
Projected area big-end bearing (square inches)	7.5	4.95	5.5	8.12

With the above dimensions the loading (cl.) on the big-end bearing, due solely to the rotating weights, will be as follows:—

(a) 200 pounds square inch.
(b) 433 pounds square inch.
(c) 495 pounds square inch.
(d) 682 pounds square inch.

The rubbing velocity (rv) is given by the formula:—

$$rv = 0.00436 \times d \times n \text{ (feet second)}.$$

d being the diameter of the crankpin (inches).

The load factor (lf) for the centrifugal loading only, will be equal to the centrifugal loading (pounds square inch) multiplied by the rubbing velocity (feet second).

$$lf = cl \times rv \text{ (pounds foot second)}.*$$

* The Michell principle does not follow this rule.

With the above assumptions we get the following values for the load factor:—

(*a*) $200 \times 18.85 = 3{,}770$ pounds foot second.
(*b*) $433 \times 20.1 = 8{,}700$ pounds foot second.
(*c*) $495 \times 20.5 = 10{,}150$ pounds foot second.
(*d*) $632 \times 22.4 = 14{,}150$ pounds foot second.

In considering these figures a comparison can only be usefully made between engines (*a*), (*b*) and (*c*) as in these three types it is customary to use the same type of big-end bearing (*i.e.*, white metal on steel), whereas in the case of radial engines the big-end bearing is usually fitted with ball or roller bearings. On the other hand the use of ball or roller bearings very considerably increases the rotating weight so that full advantage cannot be taken of the fact that they can be more heavily loaded with safety than a plain bearing. It would appear that a hopeful solution for the big-end bearing of a radial engine would lie with a composite plain bearing, consisting of a number of floating bushes with adjacent rubbing surfaces of case-hardened steel and bronze.

Referring again to the four types of engines under consideration, the following table shows the variation in revolutions per minute piston speed, centrifugal loading and load factor under various conditions:—

Permissible Increase in Engine Speed for a Constant Centrifugal Loading on Big-End Bearing.

Type of engine	(*a*)	(*b*)	(*c*)	(*d*)
Revolutions per minute for piston speed = 2,000 feet/minute	1,750	2,140	2,140	1,960
Centrifugal loading on big-end bearing (pounds square inch)	200	433	495	632
Permissible increase in engine speed for centrifugal loading = 632 pounds square inch, per cent	78	21	13	0

Permissible Increase in Engine Speed for a Constant Load Factor—(centrifugal loading only).

Type of engine	(*a*)	(*b*)	(*c*)	(*d*)
Revolutions per minute for piston speed = 2,000 feet/minute	1,715	2,140	2 140	1,960
Load factor—pounds feet second	3,770	8,700	10,150	14,150
Permissible increase in engine speed for load factor = 14,150 pounds foot second, per cent	55.5	18	12	0

Values of centrifugal loading and load factor for constant speed = 2,000 revolutions per minute. This condition is that of constant power if the value of p is assumed to be the same for all engines.

Type of engine	(*a*)	(*b*)	(*c*)	(*d*)
Centrifugal loading on big-end bearing (pounds square inch)	272	378	432	660
Load factor (pounds feet second)	5,980	7,120	8,300	15,100

By far the most heavily loaded bearing in six, eight or twelve-cylinder engines or, in fact, in any arrangement other than the radial type, is the center crankshaft journal bearing. The loading on this bearing can, of course, be reduced by the use of balance weights, but this method of dealing with the trouble entails other disadvantages apart from the considerable increase in the weight of the engine.

In the author's opinion, as just stated, however, the loading on this bearing can be dealt with by the employment of roller bearings, or, perhaps, special sleeve types, or Michell bearings.

It will be seen that the centrifugal load varies as the square of the speed, whereas the value of the load factor varies as the cube, so that any reduction in the rubbing velocity by the use of floating bushes, for example, would be of considerable assistance quite apart from their tendency to reduce local wear on the crankpin.

TABLE 17.

BEARING LOADINGS AND LOAD FACTORS FOR BIG END AND CENTRE JOURNAL BEARINGS. (SIX CYLINDER VERTICAL ENGINES.)

Engine.	R.P.M.	Piston speed (Ft./Sec.)	Big End Bearing.		Centre Journal Bearing.	
			Mean average loading (lbs.sq.in.)	Load factor (lbs.ft.sec.)	Mean average loading (lbs.sq.in.)	Load Factor (lbs.ft.sec.)
150 H.P. Ricardo Tank Engine	1200	1500	421	6330	400	6020
220 H.P. Benz	1400	1745	421	6070	805	12000
260 H.P. Mercedes ..	1400	1654	440	6780	595	9040
300 H.P. Maybach ..	1400	1654	580	9200	753	11920

NOTE.—The load factor for centre journal bearing in Benz and Maybach engines is unusually high for German practice, and it is interesting to note that these two engines are the only German engines fitted with ventilating arrangements to the crankcase, and in the case of the Benz engine, a form of oil cooler is incorporated in the sump.

In addition to the loading imposed solely by the rotating masses, the big-end bearing is also subjected to loading due to the fluid and inertia pressures. The mean fluid pressure is approximately constant and for the purpose of comparison may be taken as being equal to 50 pounds square inch multiplied by the total area of the pistons attached to the crankpin, multiplied by a constant depending on the grouping of the cylinders.

The mean average loading due to inertia will vary in the different types of engines according to the arrangement of the cylinders.

With regard to the permissible value of the load factor for the big-end bearing, for continuous running, it is of interest to note that Güldner in "Design and Construction of Internal Combustion Engines" gives 1,650 pounds feet seconds as a maximum value "*where very careful construction and perfect lubrication is assured.*" Güldner is, however, only referring to relatively large slow speed engines, where rather different conditions prevail as regards lubrication and to engines which are required to run continuously for very long periods without overhaul.

In view of Güldner's figure, quoted above, it is of interest to note the very high values of the load factors, attained in several different types of engines which have proved successful in service.

The following figures have recently been published by the U. S. Government regarding the 12-cylinder Vee type Liberty aircraft engine. The figures given are for the total loading, fluid, inertia and centrifugal.

LIBERTY 12-CYLINDER VEE AIRCRAFT ENGINE.

Bearing loadings and load factors, 1,700 revolutions per minute.

Big-end Bearing:		
Centrifugal loading only	lbs. sq. in.	366
Total average loading	lbs. sq. in.	642
Rubbing velocity	ft./sec.	17.7
Load factor (Total average loading)	lbs. ft. sec.	13,500
Center journal bearing:		
Total average loading	lbs. sq. in.	1,265
Rubbing velocity	ft./sec.	19.5
Load factor	lbs. ft. sec.	24,670
Intermediate journal bearings:		
Total average loading	lbs. sq. in.	700
Rubbing velocity	ft./sec.	19.5
Load factor	lbs. ft. sec.	13,650
End journal bearing:		
Total average loading	lbs. sq. in.	610
Rubbing velocity	ft./sec.	19.5
Load factor	lbs. ft. sec.	11,900

From the figures given for the Liberty engine, it would appear that it is actually possible to employ a load factor, as high as 24,000 pounds feet seconds continuously, provided that very ample lubrication is assured, together with an efficient system of oil cooling, but there is little doubt that with such a high value, wear will take place, and in order to reduce such wear to a minimum, it would be necessary to employ very hard steels for the crankshaft. It should also be pointed out that in such high speed engines the wear on the crankpin is very local.

It is interesting to note that the specific bearing pressure depends solely on the projected area of the bearing, whereas the load factor is influenced only by the length of the bearing.

For example, take the case of the center journal bearing of the Liberty aircraft engine when running at 1,700 revolutions per minute.

Diameter of journal	= 2.625 inches.
Effective length journal bearing	= 1.75 inches.
Projected area	= 4.59 square inches.
Total average loading	= 5,810 pounds.
Mean average loading (5,810/4.59)	= 1,265 pounds square inches.
Rubbing velocity	= 19.5 feet/seconds.
Load factor (1,265 × 19.5)	= 24,700 pounds feet seconds.

Assume diameter of the crankshaft to be increased from 2.625 inches to 3.00 inches:

Projected area of bearing	= 5.25 square inches.
Mean average loading (5,810/5.25)	= 1,110 pounds square inches.
Rubbing velocity	= 22.3 feet/seconds.
Load factor (1,110 × 22.3)	= 24,700 pounds feet seconds.

i.e., any increase in shaft diameter is neutralized by the increase in the rubbing velocity.

Taking now the diameter of the crankshaft as in the first case, *i.e.*, 2.625 inches and lengthening the bearing from 1.75 inches to 2.25 inches.

Projected area of bearing (2.625 × 2.25)	= 5.9 square inches.
Mean average loading (5,810/5.9)	= 985 pounds square inches.
Rubbing velocity	= 19.5 feet/seconds.
Load factor (985 × 19.5)	= 19,200 pounds feet seconds.

Taking the limit type engine, the straight six, with an allowable increase of speed of 50 per cent, the weight comes out approximately the same as the radial.

The loading factor of the big-end in the radial may be reduced either by giving the cylinders an opposite rotation to the crankshaft, as in the differential, thus making possible an increase of speed of about 20 per cent, or by the adoption of special bearings, or other means not yet known.

New mechanical arrangements of cylinders, such as that employed in the swashplate type, may give advantages in this respect, though they will undoubtedly set up many other mechanical problems that will take time to solve satisfactorily.

With regard to valve gear, high volumetric efficiency involves the use of ample valve area, and large valves give rise to trouble in their operation at high speeds. In stationary vertical and Vee type engines the weights of reciprocating parts of the valve gear may be reduced by the use of overhead camshafts operating directly on the ends of the valve spindle as in the Hispano-Suiza design.

In radial engines the valves must be operated by cams at a considerable distance from the cylinder heads. It is obvious that for the higher speeds the weight must be still further cut down; the shock of reversal and infinite acceleration due to over clearance are further difficulties; unistressed gears, *i.e.*, always stressed in one direction, and the use of centrifugal force reduce weight and obviate the use of heavy springs. If the cylinders are made to rotate at a comparatively low speed, a light tension valve gear stressed always in one direction by centrifugal force may be employed and with practically no valve springs.

As an example, let us take the two cases of an engine 6.00-inch bore and 7.00-inch stroke running at 1,500 revolutions per minute and 2,500 revolutions per minute. We will assume that we wish to employ a mean gas velocity of 150 feet per second in both cases.

Let A = Area piston, square inches.
K = Piston speed, feet seconds.
d = Diam. valve port, inches.
h = Lift of valve, inches.

We will further assume that the area through the valve (a) is equal to
$a = 3.14\ dh$ (square inches).
Then for the required gas velocity
$a = AK/150$ (square inches).
Solving for the two engine speeds, we have

(1) $a = 28.3 \times 29.2/150.$
$= 5.5$ square inches at 1,500 revolutions per minute.

(2) $a = 28.3 \times 48.6/150.$
$= 9.16$ square inches at 2,500 revolutions per minute.

This means that for 1,500 revolutions per minute we shall require four valves 1.87 inches diameter and for 2,500 revolutions per minute four valves 2.42 inches diameter.

The relative weights of the valves will be as

1.87^3	:	2.42^3
$= 6.55$	:	14.2
$= 1$	:	2.16

and the required lift (h)

$h = 0.4675$ inch	:	0.605 inch
$= 1$	:	1.29

The difficulty of operating the valve gear of an engine of this size at such a high speed is at once apparent from the above figures.

Apart from other limiting factors it is also obvious that a limit will eventually be reached when the revolution speed is such that the connecting rod will fail in bending due to its own inertia.

With regard to the rate of heat flow, the advantage undoubtedly lies with the small cylinder both as regards the uncooled parts of the engine, such as pistons, valves, plugs, etc., and also in the relative size of radiator required.

Instead of poppet valves involving rapid reversal of direction of motion, rotating valves constantly moving in one direction may be employed, for example, sleeve valves. This type, with an additional reciprocating motion, was used in the Argyll engine which competed in the aeroplane engine trials in 1914, but very little has been done since, and knowledge as to its probable behavior for aero work is lacking.

Considerable experiment would probably be required to attain successful results. The fundamental difficulty seems to be that of cooling, particularly transferring the heat through the oil film between the sleeve and the cylinder.

CYLINDERS.

In the consideration as to the trend of cylinder design in the direction of either water or air-cooling, the tests carried out at Farnborough will have a very complete bearing.

This research is of the greatest value in connection with the development of the radial engine, both with fixed and rotating cylinders, and has definitely established that, with suitable materials and design, mean effective pressures can be obtained in air-cooled engines nearly as high as those which have been obtained with water-cooling, and with consumptions reduced to a figure approaching the best hitherto obtained in water-cooled engines.

Experience with the R.A.F. 4 D cylinder, and modified cylinders, led to the design of the 21 T.D. cylinder, and later the smaller 22 T.W. cylinder. Both have aluminum heads cast on to steel barrels. These cylinders were very successful, as will be seen from the particulars given in Table 18, which gives their performance arranged as single cylinder units.

TABLE 18.

RESULTS OF TESTS ON AIR COOLED CYLINDERS.

Type of Cylinder.		Dragonfly Mk. II.		R.A.E. 21 T.D.		R.A.E. 22 T.W.		R.A.E. 22 T.W.	
Bore of Cylinder	Inches.	5.5	5.5	5.5	5.5	4.5	4.5	4.5	4.5
Stroke of Piston	Inches.	6.5	6.5	6.5	6.5	6	6	6	6
Revs. per Minute	R.P.M.	1450	1650	1450	1650	1800	2000	1800	2000
Brake Horse Power	B.H.P.	31.4	34.0	36.0	39.7	29.0	31.08	28.9	32.35
Brake Mean Pressure	B.M.E.P.	110.2	105.1	127.0	123.0	133.9	128.8	133.6	134.0
Fuel Consumption	Lbs. B.H.P. Hr.	0.722	0.793	0.518	0.537	0.430	0.535	0.468	0.477
Weight of Cylinder	Lbs.	24.25	24.25	26.88	26.88	18.4	18.4	18.4	18.4
Compression Ratio	R.	4.6	4.6	5.0	5.0	5.1	5.1	5.5	5.5

The results of these experiments which were carried out in the earlier stages by Dr. A. H. Gibson and later by Major G. H. Norman, are of the greatest value. Norman's conclusions as to the present position of

development of the air-cooled engine without taking account of future experiments, are stated as follows. That an air-cooled engine can be built:—

(1) Having a B.M.E.P. of 120 pounds per square inch on the complete engine.
(2) Of weight not more than 2 pounds per B.H.P.
(3) Capable of standing full load under the same conditions as a water-cooled engine.
(4) Having a fuel consumption not exceeding 0.55 pound per B.H.P. hour.

TWO-CYCLE ENGINES.

Though the two-cycle engine has always had a great attraction, particularly to the non-technical, it has never made much progress for aero engine work. No doubt the fact that it had twice the number of working strokes of the engine ordinarily used, the four-stroke, was the reason of this. It was further supposed to be simpler. In these comparisons the charging pumps with their valves were somewhat overlooked, but the real difficulty and the one to keep the engine out of use was its low fuel economy, and the difficulties of getting rid of the extra heat.

The charging of the working cylinder with mixture was the main cause. Some of it escaped, and some inert gas remained in the cylinder to heat up the rest, reduce its efficiency and render it more liable to pre-ignition. Proper scavenging at high speed is extremely difficult.

The advent of the direct injection method has greatly changed this situation, and there now seem much greater possibilities of the two-cycle coming into use for aero work, and probably land transport as well.

The direct injection method of Akroyd Stuart of supplying fuel to the engine cylinder has realized its present efficiency through its development in the Diesel and high compression oil engines, and it is dealt with elsewhere. By its use the engine cylinder can now be scavenged by air alone, the fuel being injected direct in the constant volume cycle, just as the ports are closing or immediately after. Thus the old fundamental trouble of escape of mixture is overcome. At the present time the two-cycle principle is in use in Diesel and high compression engines.

For aero work when large port area is essential, piston type valves are most likely to be employed. This type of valve with its regular action and low acceleration stresses allows of high speeds. Thus we may get a two-cycle engine running at higher speeds than a poppet valve four-cycle with safety.

With regard to the increase of power, it would not be safe to count on more than 50 per cent over that of a four-cycle of equivalent cylinder capacity.

This means that 50 per cent of additional heat has to be dissipated in the same time. Hence, in the first development water-cooling is likely to be adopted, though ultimately it may be possible to use air.

The scavenging and charging of the cylinder may be carried out by turbo-blowers worked from the exhaust pressure as in the forced induction trials on the four-cycle.

Economy of fuel will not be so good as with the four-cycle in the first instance, still 0.6 pound should be achieved, and with the reduction in weight that will be possible, should, for flights under three hours, make the type quite suitable. The loadings on the big-end will be, in effect, worse, hence some compensating device may be necessary if the type chosen is the radial.

For civil aviation at present, only well-known and tried types are likely to be employed, and the choice of practicable engines at the moment appears to be between the simple six-cylinder in line, multicylinder Vee and double Vee engines, though the first must reduce its weight ratio by raising the rotational speed, when the difficult problem of gear reduction to allow efficient propeller speed will come in. In the author's view, these are the types at the moment whose reliability renders them suitable for commercial work; civil aviation will ultimately depend for success on the development in performance and reliability of lighter types which have been discussed.

With regard to metallurgical development, the possibilities of improvement are mainly in change in the materials of construction.

During the war, higher and higher value steels were brought out by the metallurgists allowing stresses to be put up and weight of parts cut down. Also the use of aluminum alloys reduced the weight of cast parts, as the pistons, crank cases, etc., and greatly improved the conductivity of heat-carrying parts.

Thus the advance went on parallel lines, the higher value steel allowed higher speeds, while the greater rate of heat flow was met by the use of aluminum.

There is no doubt that progress will continue in these ways from the further improvement of the materials. Possibly there may be more radical changes, the substitution of a metal of low specific gravity for steel.

The table of specific gravities already shown is interesting. The German alloy electron has a specific gravity of 1.8, but is not included in the Table.

For airship service engines are required to run continuously for long periods, and for these long runs weight of fuel and oil becomes of considerable importance as compared with engine weight. Further, since non-stop runs of 100 hours or more must be contemplated, the reliability of the motor must be of a high order.

Accessibility is also deemed essential to enable examination and adjustment and replacement of parts in flight and to enable minor repairs to be made.

At present the engines which have been employed are very similar to the ordinary vertical and Vee type aeroplane engines.

There is undoubtedly room for considerable advance in engines for airships without departing from the four-stroke cycle and by adopting modifications of the cycle by which increased economy can be obtained. The method of stratification described would appear to be particularly suitable for employment, owing to its very high economy, particularly in cases, common in airship work, where only a part of the full power is required for long periods. The great advantage of economy will be apparent from an example; taking an engine of 300 H.P., with weight 1,500 pounds and consumption 0.5 pound per B.H.P. hour. For a 100 hours' run the fuel required would be 15,000 pounds, or ten times the engine weight. A reduction of fuel consumption to 0.45 pound would reduce the weight of fuel by 1,500 pounds, so that double the engine weight would be allowed. Consumption considerably lower than this can be attained, namely, in the region of 0.4 pound per B.H.P. hour.

In the high compression direct injection method with a fuel consumption of 0.4 pound per B.H.P. hour the consumption would be 3,000 pounds less. This type will, however, require to be made lighter, possibly of the opposed piston form, before it can compete with the most economical forms of the constant volume engine for aero work.

Many proposals have been made for the use of steam turbines for aero work, but steam turbine efficiencies can by no means compete with those obtainable by the internal combustion method, as indicated in an earlier lecture, particularly in units of the low power output required.

The feature of being able to use crude oil is a valuable one, but can be secured by the use of high compression internal combustion engines with greatly reduced fuel consumption.

There are undoubtedly future possibilities in the application of the internal combustion turbine to aero work, but the gas turbine for any purpose is in a very immature stage of development, and consideration of its relative advantages would seem to be premature.

In view of the attention which it is hoped will be directed to commercial aviation, it may be of interest to consider the effect of the weight of the engine on the carrying capacity of a given machine. It will, of course, be obvious that for short flights where little fuel has to be carried and the weight of fuel is small relatively to the weight of the engine, the use of an engine of low weight per horsepower is of fundamental importance, and economy is comparatively unimportant. It is also obvious

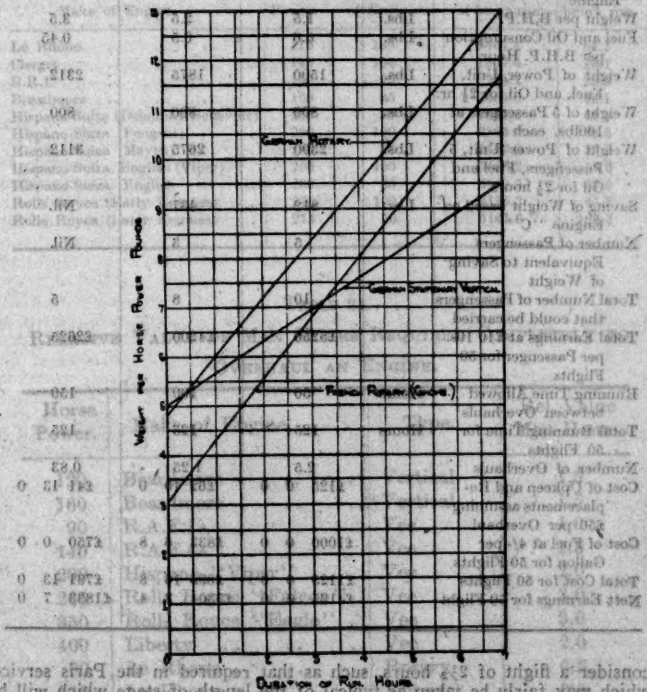

FIG. 19.

that for very long flights where the weight of fuel to be carried becomes large relatively to the engine weight, low weight of engine is of less importance than economy of fuel and oil consumption. This is very clearly shown by the curves, figure 19, which in the abscissae represent hours run and the ordinates weight of engine and fuel per horsepower in pounds.

The performances of a number of different engines are shown, and the heavier early German engines are compared with the light Gnome engine. It will be seen that for runs of anything over three hours, the average German engine has the advantage, while below three hours run the light engine gives an undoubted saving in weight and consequent greater carrying capacity, and would, therefore, seem a better commercial proposition. As an example of the commercial importance of a low weight engine, we may

TABLE 19.

LONDON-PARIS SERVICE.

Type of Engine.		a	b	c
Brake Horse Power of Engine	B.H.P.	500	500	500
Weight per B.H.P.	Lbs.	1.5	2.5	3.5
Fuel and Oil Consumption per B.H.P. Hour	Lbs.	0.6	0.5	0.45
Weight of Power Unit, Fuel, and Oil for 2½ hr.	Lbs.	1500	1875	2312
Weight of 5 Passengers at 160lbs. each	Lbs.	800	800	800
Weight of Power Unit, 5 Passengers, Fuel and Oil for 2½ hours	Lbs.	2300	2675	3112
Saving of Weight based on Engine "C"	Lbs.	812	437	Nil.
Number of Passengers Equivalent to Saving of Weight		5	3	Nil.
Total Number of Passengers that could be carried		10	8	5
Total Earnings at £10 10s. per Passenger for 50 Flights		£5250	£4200	£2625
Running Time Allowed between Overhauls	Hours	50	100	150
Total Running Time for 50 Flights	Hours	125	125	125
Number of Overhauls		2.5	1.25	0.83
Cost of Upkeep and Replacements assuming £50 per Overhaul		£125 0 0	£62 10 0	£41 13 0
Cost of Fuel at 4/- per Gallon for 50 Flights		£1000 0 0	£833 6 8	£750 0 0
Total Cost for 50 Flights		£1125 0 0	£895 16 8	£791 13 0
Nett Earnings for 50 Flight		£4125 0 0	£3304 3 4	£1833 7 0

consider a flight of 2½ hours, such as that required in the Paris service, which may fairly be taken as typical of the length of stage which will be required in commercial aviation. This length of flight would seem to be the most usual in charge of a single pilot.

Assuming power units of 500 B.H.P. with weights per B.H.P. 1.5, 2.5, and 3.5 pounds, with fuel and oil consumptions of 0.6, 0.5, and 0.45 pound respectively, the weight of power unit and fuel in the three cases would be:—

(*a*) 1,500 pounds. (*b*) 1,875 pounds. (*c*) 2,312 pounds. With the lightest engine (*a*), therefore, the saving in weight would be 812 pounds equivalent to five passengers, and with the medium weight engine, 437 pounds, equivalent to three passengers.

The earnings of fifty London-Paris flights at ten guineas per passenger are given in the following Table.

It will be seen from Table 19, that the total net earnings in the case of the lightest engine will be £4,125, in the case of the medium engine £3,304 3s. 4d., and in the case of the heavy engine £1,833 7s.

TABLE 20.

AVERAGE NUMBER OF HOURS RUN BETWEEN OVERHAULS.

Make of Engine.	Horse Power.	Total number of Engines.	Total number of hours run.	Average of hours run.
Le Rhone	110	100	5407.4	54.1
Clerget	140	100	1507.4	15.1
B.R.1.	150	25	734.7	29.5
Beardmore	160	35	1685.0	48.1
Hispano-Suiza (Delaunay Belleville)	200	25	1037.5	41.5
Hispano-Suiza (Peugeot)	200	100	3864.6	38.6
Hispano-Suiza (Mayen)	200	84	2492.2	29.7
Hispano-Suiza, English (Viper) ..	200	100	5466.2	54.6
Hispano-Suiza, English	200	93	3449.5	37.0
Rolls Royce (Early Engines) ..	275	100	6135.3	61.3
Rolls Royce (Later Engines) ..	275	50	5161.6	103.2

TABLE 21.

RELATIVE VALUE OF MAN HOURS REQUIRED COMPLETELY TO OVERHAUL AN ENGINE.

Horse Power.	Make of Engine.	Type.	Rel. Value Man-Hours.
120	Beardmore	Vertical	1.5
160	Beardmore	Vertical	1.5
90	R.A.F.1a	Vee	1.0
140	R.A.F.4a	Vee	1.5
200	Hispano "Viper" ..	Vee	2.0
265	Rolls Royce "Falcon" ..	Vee	3.0
350	Rolls Royce "Eagle" ..	Vee	3.0
400	Liberty	Vee	2.0
80	Le Rhone	Rotary	0.5
110	Le Rhone	Rotary	0.5
100	Mono Gnome	Rotary	1.0
130	Clerget	Rotary	1.0
200	B.R.2	Rotary	1.0

The earnings for a given carrying capacity, it will be noted, are very large in proportion to any fuel costs likely to be incurred even in the most wasteful of engines as regards consumption.

In this calculation, no account has been taken of the initial cost, as this is independent of type, and with greater development may well be in favor of the lighter engine. The provision and maintenance of a given number of engines in working order is allowed for in the renewals and replacement item. These figures very clearly show that for flights, such as those required on the London-Paris services, light weight per H.P. is of the first importance.

It has been assumed that the consumption in the light weight engine is very much greater than that in the heavier engine. With further development of the lighter types this is by no means necessary, and improvements in fuel consumption would obviously increase the balance in favor of light weight.

The above Tables, numbers 20 and 21, have been compiled by Lieut.-Col. L. F. R. Fell, D.S.O., from the records of the engine repair shops of the Royal Air Force in France, and are included on account of their great interest. Table 20 is a summary showing the average number of hours run between overhauls of various engines under active service conditions. It is of particular interest in that the figures for the 200 horsepower Hispano-Suiza indicate very clearly the effect of manufacture on the same type of engine. In the case of the Rolls Royce, two independent batches and averages have been taken. With regard to Table 21, showing the relative value of the man-hours required completely to overhaul, the short period of time necessary for the simple type radial as seen in the case of the Le Rhone engines should be noted.—"Journal of the Royal Society of Arts."

RECENT PROGRESS IN LARGE DIESEL ENGINES FOR THE MERCANTILE MARINE.

By James Richardson, B.Sc., Asso.M.Inst.C.E.

Prior to the war, with the exception of a few four-stroke cycle engines of relatively small power, little real progress was made in the application of Diesel engines to ships in the mercantile marine.

During the years 1914-19, however, a steady and consistent policy of development took place, naturally, under the circumstances, mainly confined to neutral countries. The four-stroke cycle single-acting, crosshead type of engine, with forced lubrication, went through a process of gradual evolution. There was no radical change either in thermodynamical theory, in practical application, or in the metals and materials used; the efficiency, moreover, remained almost constant. With increasing power and growing experience, the cylinder mean effective pressure considered desirable for continuous running was gradually reduced from 85 pounds to 75 pounds per square inch on a brake horsepower basis. A number of ships were engined with two-stroke cycle Diesel machinery, but the progress made was less definite and less convincing.

From 1912 to the present time, the horsepower per single cylinder has risen by small steps from little more than 100 B.H.P. to over 300 B.H.P., and, with multiplication of cylinders, a maximum power of approximately 4,800 B.H.P. (with 16 cylinders on twin screws) per ship can be obtained with machinery of proved design.

Since the war, conditions the world over, in all spheres of activity, have been unique. Difficulties with all classes of labor ashore and afloat, increased transporting tonnage now available for liquid fuels, and the recent substantial fall in price, have compelled attention to the subject of the substitution of oil for coal. The economic advances, under present conditions, of the liquid over the solid combustible for power generation, have been quickly and generally realized.

The Diesel oil engine utilizes oil fuel with an economy far exceeding that of all other prime movers, and the total savings due to the adoption of the principle of internal combustion are even greater than is solely consequent upon the much lower consumption of fuel of from one-third to one-fifth of that required for equally powered steamers, since, apart from fuel costs, there are a number of other important gains.

All shipowners are now, at least, giving close attention to the merits of this system of propulsion, and construction is advancing. The gross tonnage of ships to be equipped with Diesel machinery launched in 1913 was 60,000 tons. In 1919 the figure was 85,000, in 1920, 189,977, whereas building last year there were 454,502 tons of Diesel-driven ships, or over 7 per cent of the world's total under construction. In the light of the past ten years experience, the statement will not be denied that the Diesel engine has proved an efficient, reliable and thoroughly seaworthy prime mover, suitable for a large proportion of the total sea-borne tonnage. In 1920, of 364 ships launched of more than 2,000 gross tons each, 350, or 96 per cent of the total number, required less than 5,500 I.H.P. of machinery per ship. In numbers 96 per cent, and in tonnage 88 per cent, come thus within the scope of the oil engine, so far as total horsepower is concerned.

The principles underlying the operation of internal-combustion machinery are now becoming more widely understood, at least so far as the simple applications are concerned, making easier the path of the advocate of this form of prime mover.

The technical pride of the engineer, and the desire on the part of the shipowner for greater economy with the larger type of ships, such as intermediate liners, will not allow the problem to remain long unsolved for powers higher than 4,800 B.H.P. per ship. In the near future the maximum power with oil engines will increase. This problem is being attacked in this country, and the large variety of types under construction here indicates an enterprising and progressive policy of determined trial, in contrast with the apathy with which British engineers were charged before the war.

The rival claims of some of these various systems indicated in the table have yet to be fully sifted in the light of practical sea-going experience. It is most probable, however, that a considerable time will elapse before development and increased application will enable a substantially higher plane of performance and efficiency to be reached, when the issue between the various types will become narrowed down. It is hoped that the results of sea-going experience will be freely disseminated and that the policy of secrecy, which is to be condemned and which, at least in its initial stages, particularly characterized this movement, will not be pursued. The power which various types of engines can consistently and continuously sustain at sea is more particularly a point in question, about which little information can generally be gleaned.

The maximum power which can be obtained from a given cylinder depends primarily on the limits of heat transfer from the working fluid through the metal surrounding the combustion space to the cooling medium. Attention is directed to the last column in the table indicating this quantity conveniently expressed in terms of pounds of fuel per square inch of piston area, showing the divergence in published ratings in this

List of Important Diesel Ships in Service and Building Giving Particulars of Their Machinery.

Date.	Name of Vessel.	Makers of Machinery.	Type of Engine.	Cycle.	I.H.P. of Engine.	B.H.P. of Engine.	No. of Cylinder	B.H.P. per Cylinder.	Diameter of Cylinder in In.	Stroke, In.	Ratio Stroke to Bore.	R.P.M.	Piston Speed, Ft. per Min.	M.E.P. on B.H.P. Basis.	M.I.P. on I.H.P. Basis.	Consumption of Fuel Lb. per Sq. In. Piston Area.
1912*	Juno	Werkspoor	Werkspoor	4 single-act.	1,400	1,100	6	183	21½	30½	1·87	125	820	81·0	111·0	0·219
1919*	Glenapp	Harland and Wolff	Burmeister and Wain	4 single-act.	3,200	2,625	8	328	29½	43⅞	1·46	115	830	76·0	93·0	0·202
1920*	Afrika	Burmeister and Wain	Burmeister and Wain	4 single-act.	2,250	1,700	6	282	29½	45½	1·55	115	865	65·0	86·0	0·179
1920*	Glenogie	Harland and Wolff	Burmeister and Wain	4 single-act.	3,200	2,625	8	328	29½	45½	1·55	115	865	75·0	91·5	0·206
1920*	Cuhore	Bethlehem Steel Co.	Bethlehem	2 single-act.	3,900	2,700	6	450	25½	48	1·87	100	800	74·0	105·0	0·39
1920*	Ansaldo San Giorgio III	Ansaldo San Giorgio	Ansaldo	2 single-act.	1,900	1,200	4	300	24⅛	35⅛	1·43	110	650	63·0	84·0	0·272
1920*	Fullagar	Cammell Laird	Fullagar	2 opposed p.	660	500	4	125	14	20	1·43	110	367	73·5	97·0	0·179
1920*	Fritz	Blohm and Voss	M.A.N.	2 double-act.	1,250	950	3	263	18½	28	1·48	110	515	65·0	93·5	0·223
1920*	—	Swan - Hunter and Wigham Richardson	Polar	2 single-act.	1,500	1,130	6	188	18	37	2·06	115	710	69·0	92·0	0·325
1920	—	Gotaverken	Burmeister and Wain	4 single-act.	1,600	1,200	6	200	25	51	2·04	95	805	67·0	89·5	0·171
1921	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
1921	Douala	N.B. Diesel Co	N.B. Diesel	4 single-act.	2,336	2,000	8	250	26½	47	1·77	98	750	79·5†	92·0	0·191
1921*	Sardinia	Werkspoor	Werkspoor	4 single-act.	2,140	1,600	6	266	26½	47½	1·79	100	788	82·6	110·0	0·204
1921*	Seminole	Vickers, Ltd.	Vickers	4 single-act.	1,620	1,250	6	208	24	39	1·59	118	767	76·0	99·0	0·185
1921*	Pinzon	Beardmore and Co.	Beardmore-Tosi	4 single-act.	1,600	1,200	6	200	24	38½	1·57	120	767	74·0	98·0	0·180
1921	—	Stephen and Sons	Sulzer	2 single-act.	2,280	1,650	4	412	26½	43⅞	1·62	85	610	78·0	106·0	0·321
1921	—	Denny Brothers	Sulzer	2 single-act.	1,700	1,250	4	312	23½	37	1·56	100	615	76·0	103·0	0·314
1921	—	Cammell Laird	Fullagar	2 opposed p.	1,580	1,200	4	300	18½	25	1·33	115	480	77·0	101·0	0·248
1921	—	—	—	2 opposed p.	4,000	2,700	4	675	32½	45½	2·0	77	585	93·0	103·0	0·362
1921	Yagaron	Doxford	Doxford	2 opposed p.	4,000	2,700	4	675	32½	45½	2·0	77	585	93·0	103·0	0·362
1921	Suphonco	James Craig Engine Works, U.S.A.	Craig	4 single-act.	2,240	1,850	6	308	30	48	1·6	105	840	68·5	82·0	0·183

* In operation at sea.

† No compressors on main engine.

respect, although it is suggested by the author that the results in practice, so far, do not differ so widely, and that the quality and rate of heat transfer continuously and successfully achieved, are much nearer the lower than the higher figure. The application of metallurgical research and improved methods of construction are the paths being traversed towards the higher results desired.

Another basis of comparison concerns the qualities of fuel which various types of engines can consume economically in respect of the amount of attention to be given to the running parts. The types of fuel oils now running on the market are showing a tendency to deteriorate in quality, increasing in viscosity, and decreasing in calorific value, so that the engine designer, constructor and operator must continue to study their efficient utilization.

Various designs and types of engines, whether working on the two or four-stroke cycle principle, single or double-acting, or utilizing one or other of the alternative systems of fuel feeding, will, therefore, be judged largely on the capacity to burn satisfactorily and economically various qualities of oil fuel, including lower grade oils, with minimum attention and upkeep costs on a basis of horsepower continuously developed at sea.—"Engineering," July 22, 1921.

ADVANTAGES OF THE DIESEL-ELECTRIC DRIVE APPLIED TO MARINE PROPULSION.

To the Editor of "Motorship":

In a recent paper (by Messrs. J. F. Metten and J. C. Shaw) which has attracted considerable attention, the Diesel-Electric Drive System was referred to in rather unfavorable terms. Like the majority of other questions, there are two sides to it, and the ship-owner who contemplates the use of Diesel engines—and there are very many who are always in the van of progress, seriously taking thought of the morrow—should and would carefully consider all the factors before making a decision.

In the selection of the motive power for any vessel, various factors enter into the problem, in about the following order:

1st. First cost.
2nd. Suitability for the required type of ship.
3rd. Space and weight occupied.
4th. Cost of operation.
5th. Reliability.
6th. Life of the installation.

If it were possible to assign numerical values to each of the pros and cons affecting each of the factors it would then be possible to arrive at a definite conclusion free from prejudice and opinion. But unfortunately these factors cannot be so valued excepting where certain definite propositions are put up, one against the other, and where actual experience and data are available. It is to be regretted that at the present time sufficient experience is not available, so the problem reduces itself to a comparison of actual bids and guarantees, together with the exercise of considered judgment and common sense toward any factors not covered by such bids and guarantees.

With the data and experience that has been obtained the following statements may be made relative to the several factors.

First Cost.—The most direct answer to this question is to get bids from the builders of the direct-drive and the electric-drive systems. The New London Ship and Engine Company, whilst ready to bid and build on both systems, has worked in conjunction with manufacturers of electric equipment and is prepared to make bids on the complete electric-drive system. Its estimates and experience to date indicate a slightly cheaper first cost for the electric drive installation. There are two reasons contributing to the above result, one of which is that the total weight of such an installation is less, and consequently the cost of material is less, and the other is that both the engines and electrical machinery are standard, and so can be manufactured in quantity, thus taking advantage of the well-known saving due to mass production.

A further reason is that it is not necessary for the auxiliary generating units in the Electric-Drive installation to be so numerous or so large as in the case of the direct-drive, for when handling cargo, assuming electric power is utilized as is the case of practically all Diesel installations, one unit of the main engine is used, efficiently and economically, while one, two, or three auxiliary units of fairly large capacity are required for this work when the main engines are direct connected. Add to this that the provision made for supplying maneuvering air and its storage for this type is greatly in excess of that required for starting purposes only of the other type, and that this costs money, it can be readily seen that the cost of main propelling machinery for electric drive includes a large part of the cost of the auxiliary machinery, which is not so included in the first cost of a direct drive proposition.

Suitability of the Machinery for the Required Type of Ship.—The type of ship depends in many cases on the individual preference of shipowners, based for the most part on practical experience. A single-screw ship is considered essential by many, partly for the sake of higher propeller efficiency, partly because shipbuilders construct this type more cheaply than that with twin screws, partly to enable decent afterbody lines to be obtained with machinery placed aft, and partly to prevent, so far as possible, damage to propellers when coming alongside. The electric drive is ideal for any or all of the above considerations, as the main engines may be arranged without much regard to position so long as they are in the machinery space, and thus the lines may still be fine without the machinery space being too far forward. Single screws can be used without sacrificing the advantage of having two, at least, similar sets of main engines. Fortunately these same advantages can be obtained when the electric-drive system is installed amidships. At the present time there are many shipowners who would not consider Diesel propulsion, reliable though it has proved to be, unless the vessel had more than one set of machinery capable of operating the propeller. Furthermore, there are very few single-screw installations of the power required for ships of 5,000 tons and above, and for this class once again the electric drive is ideal, as it has practically no limitations as regards power with single screws.

Space and Weight.—The slow-speed direct-drive engine is exceedingly heavy—over 400 pounds per horsepower. Its over-all dimensions are necessarily large. Therefore a correspondingly large space must be devoted to it. The Electric-Drive System, being divided into a number of smaller units, can utilize spaces not available for the larger engine. The greater the amount of power required, the greater this advantage shows up. [Weights of the 10 leading American slow-speed Diesel engines were given on page 293, April issue, and they average 339 pounds per brake H.P.—*Editor.*]

Cost of Operation.—This includes many things. It is not only fuel, lubricating oils, and attendance, but in a broad way includes depreciation, repairs, maneuvering ability and reliability. An accident, or a long time under repairs would add enormously to the cost, and make a big hole in the profits at the end of a year. Many of these factors are so great that they would completely absorb a slight difference, say 15 per cent in the fuel bill. There is a tendency to over-evaluate the advantage that exists between a fuel consumption of 0.42 and 0.45 pound per B.H.P.-hour, and undervalue the claims of simplicity, ease of repairs, and possibility of obtaining any casting or part in the majority of comparatively small engineering establishments in any part of the world. Apart from these considerations, with the smaller electric-drive engines, all identical, the outfit of spare parts will not be much larger than that required for one engine, so that it is feasible to carry every conceivable part so that every conceivable emergency may be met, without actually carrying so much weight on board compared with the direct-drive larger engine supplied with normal spare part equipment that only touches the high spots, so to speak, of possible replacements.

Repairs to the large direct-drive engine, when they become necessary, are often expensive and should an accident occur in a foreign port and the necessary spare parts not be carried on board, the resulting delay might well run into months, while the electric-drive installation with a similar accident will always be able to make the repair and still operate or, if beyond available resources, can still return to the home port, limping, it is true, but getting there.

To date there have been many types of direct-driven Diesel engines put into service, but only a few have made reputations. The failures were mostly experimental in nature, the products of over-enthusiasm and misplaced confidence. The successes are the products of conservatism and care. The design was exceedingly heavy with few if any untried features and the operating engineers were carefully trained by the builders. It is to overcome these very difficulties that the electric-drive is proposed. The engines being of moderate weight, moderate speed, and simple in the extreme, should not require a very highly specialized or numerous personnel. One engineer and one oiler can give all the attention necessary to an entire group of engines, and the electrician, usually carried in Diesel-engined ships, attends to any electrical upkeep necessary.

The question of reliability is so closely connected with that of cost of operation that it has been fairly well covered in the preceding paragraph. The great reliability of the Diesel-Electric Drive comes from two principal elements, to-wit, the inherent reliability, flexibility and smoothness of performance of electric motors; and from the reliability inherent to multiple units.

Life of Installation.—So far as can now be told, there should be no appreciable difference in the life of the two systems. The wear that will and does take place has two causes, viz.: piston speed and pressure. No matter how large or small the Diesel-engine may be, these two elements remain about the same, and are the same in good design, irrespective of size within wide limits.

The ideal internal-combustion engine would be a machine, absolutely standard, enclosed in a box with a few levers projecting, so that it could be started, maneuvered and controlled under all conditions by the movements of these levers. After a certain period of operation, it would be removed bodily for refit, being replaced by one exactly similar, the original being placed into condition for a further period of service. Connect up some such machines to electric generators and let them furnish current to the main motors and the power problem

is solved. Such an ideal machine is not yet in existence, but the Diesel-Electric Drive is a step in that direction and is considerably nearer the goal than the direct-drive especially for powers of the order required for passenger and large freight vessels.

It is fitting that the firm which originated electric-drive also has the longest experience of Diesel-engine direct-drive in the United States, so that it can look upon the question quite dispassionately due to its experience of both types and is not committed to any one type to the exclusion of the other with the enthusiasm that is often born of knowledge of one type only.

Yours very truly,

THE NEW LONDON SHIP & ENGINE CO.

E. NIBBS, Marine Engineer.

Groton, Conn. —"Motorship, New York," August, 1921.

SIX-THOUSAND TONS CABLE MOTORSHIP FOR WESTERN UNION TELEGRAPH CO.

DIESEL-ELECTRIC DRIVE TO BE ADOPTED FOR PROPULSION AND CABLE-LAYING.

Specifications have just been issued to shipbuilders for one of the most interesting steel motorships ever planned. This is a cable-laying ship of 6,000 tons displacement for the Western Union Telegraph Company of New York from designs by J. W. Millard & Sons, naval architects also of New York. It is anticipated that the contract will be placed within a few weeks, unless the cost of construction is too high, in which event the vessel may be built abroad. In any event the order will be soon placed and the ship completed, as she is needed for work on the ocean.

It is very noteworthy that steam drive will not be considered for this ship, and that shipbuilders who have no Diesel-engine of their own must include in open bids the McIntosh & Seymour, Worthington, or other equivalent makes of Diesel engines that can be installed in the space avaliable, in conjunction with Westinghouse electric transmission.

Coming at a time when steamship-building is suffering from a terrific slump this vessel will do much to center additional attention from shipbuilders on the importance of Diesel-engines for propelling purposes, as there are still about a dozen important American concerns who have not yet undertaken the commercial construction of Diesel-engines, although several have been negotiating for licenses or hovering on the brink of doing something tangible in this direction, including the Federal Shipbuilding Co., Bath Iron Works, Staten Island Shipbuilding Co., Merchants Shipbuilding Co., Texas Steamship Co. (shipyard at Bath, Me.), Moore Shipbuilding Co., South Western Shipbuilding Co., American Shipbuilding Co., Great Lakes Shipbuilding Co., and Downey Shipbuilding Co.

In this particular instance Diesel-electric drive is ideal, because the vessel will have to operate very slowly, except when proceeding to her destination at sea, which may mean only about four days at full speed. At times she will have little more to do than drift around, with most of the power shut down. With this class of machinery the standby charges will be very low, and at any time she is ready for full-speed ahead or astern in ten seconds, without being obliged to maintain a head of steam.

There will be two 1,000 shaft horsepower 500 volt direct-current electric motors connected to the propeller-shafting, and these will turn at 120 revolutions per minute. Kingsbury thrust-bearings will be fitted. Current for the operation of the motors will be furnished from four 410 K.W. 250 volt direct-current compound-wound electric-generators driven by four 600 B.H.P. Diesel oil-engines.

For the auxiliaries there will be two 120 B.H.P. Diesel-engines driving two 90 K.W. 250 volt direct-current compound-wound generators at 375 revolutions per minute. These generators in addition to providing power for the auxiliaries of the ship, will provide current for two 250 volt, 1750 R.P.M. motors that will drive two 20 K.W. electric-light generators that will supply power for lights at 125 volts. They can also take current for the excitation bus. Power for lights thus will be taken from the auxiliary generators, through the motor-generator sets when at sea, and at such times as the auxiliary generators are being operated when in port. A small oil-engine driven 125 volts generator will furnish current for lights when the large auxiliary units are idle.

The general dimensions of the ship are as follows:

Displacement, tons	6,000
Gross Tonnage, tons	3,000
Power, S.H.P.	2,000
Registered Length, feet and inches	340–6
Length B.P., feet and inches	324–0
Breadth (Moulded), feet and inches	42–6
Depth (Moulded), feet and inches	25–0
Register	Lloyds 100 A1
Speed, knots	12

We note that when using fuel having a heating value of 18,000 B.T.U., the consumption at sea must not exceed 0.51 pound per *shaft* horsepower per hour. This, of course, makes any form of steam-electric drive utterly out of the question even if the owner's specifications on page 14 (clause 3) did not definitely state that the propelling-machinery is to consist of four Diesel-engines driving generators.

However, we think that even for Diesel-electric drive the margin is too close, because the Diesel-engines will be of fairly high speed, namely, 200 to 250 R.P.M. and so will have a fuel consumption of about 0.45 pound per B.H.P. hour, unless of the solid-injection type. To this an allowance must be made for the loss of power in transmission through the electrical apparatus of 12 to 15 per cent. Therefore, if a consumption at the propeller-shaft, of 0.575 pound were maintained it would be a very good result.

With direct Diesel-drive and slow-speed cross-head-engines consumption of under 0.42 pound per shaft horsepower hour could be guaranteed, but the benefits to be derived from electrical drive probably outweigh the difference in fuel-consumption in this particular case. With any form of steam-electric drive the best consumption will be 1.15 pounds and there would be many other fuel losses when laying cables, because the boilers could not be shut down or started at a moment's notice, like the Diesel-engines. Consequently, we suggest that the specifications of the machinery be changed to read a consumption of 0.60 pound per shaft horsepower hour, or else protect the Diesel-engine builders by inserting a clause to the effect that the consumption must not exceed 0.45 pound per brake horsepower hour, for they must not be held responsible for losses in the electrical transmission.

We note that complete sets of spare parts must be carried. For instance, each of the main Diesel-engines must have two sets of piston-rings for two cylinders, and one set of piston-rings for one piston of each of the two auxiliary engines. This in addition to four spare main-engine pistons and two spare auxiliary-engine pistons equipped with rings.

This will provide good business in these quiet times for piston-ring manufacturers, as it means about 56 spare rings of about 18 inches diameter and about 18 spare-rings of about 12 inches diameter.

As the four main-engines will probably have six-cylinders each and seven piston-rings per piston, and the two auxiliary-engines three cylinders with six rings per piston, it means that no fewer than 168 piston-rings of 18 inches diameter will be on the main-engines when installed, and 36 of about 12 inches diameter on the auxiliary engines. Or a grand total of 224 rings of 18 inches diameter and 54 rings of 12 inches diameter including spares. We draw attention to this because there are a number of firms specializing in the manufacture of piston-rings who do not seem to realize the extent of business in this particular field today.

On deck there will be three electric-winches of the Lidgerwood, or other approved single-drum type, and an electric anchor-windlass. The steering-gear will also be electric and will be controlled by a MacTaggart-Scott telemotor and operated by a Waterbury hydraulic variable speed-gear. A Hasties friction rudder-brake will be installed. A Sperry gyro compass, and two 24-inch Sperry searchlights will be carried. In fact, the electrical equipment throughout the ship will be very complete and will include heaters in the officers' cabins, bath-rooms and saloon, 500-watt flood-lights, fans, bells, Edison storage-batteries, log and speed indicator. Other parts of the ship will be heated by steam from an oil-fired 9 feet by 8 feet donkey-boiler at 125 pounds pressure. There will be two 2-ton refrigerating-plants of the Brunswick, York, or other approved type, electrically driven by a 250-volt D. C. motor. There also will be a Paracoil or other water-distilling plant.

When the contract for this novel and interesting motorship is placed we shall publish plans, and give further details regarding the auxiliaries and equipment.—"Motorship, New York," August, 1921.

THE FIRST MOTOR SHIP WITH DOUBLE-ACTING TWO-STROKE ENGINES.*

On the expiration of the Diesel master patents in the years 1909-10, a boom in marine motor building was anticipated by, among others, the Maschinenfabrik Augsburg-Nürnberg and Blohm and Voss. These firms accordingly jointly founded a research association for the purpose of developing the engine to the dimensions requisite for its use in warships. The double-acting two-stroke engine, it was held, could alone be expected to fulfil the object in view. It was decided to start by building small units, so that the problem might be studied with a minimum of expenditure. In order that the experiments should not be confined to the testing shop, but be extended to ascertain the suitability of this new type of engine for practical mercantile marine work, a twin-screw cargo vessel of 1863 net register tonnage was built by Blohm and Voss, to be fitted with engines of the class in view. The first set of engines constructed for this vessel was not installed on board, cracks having formed in the cylinders during the shop tests. The engines were accordingly used for experimental purposes, and to gain experience for the construction of the next set.

The new engines had three working cylinders of 480 mm. diameter and 710 mm. stroke, and developed 830 brake horsepower at 120 revolutions per minute. The injection compressor was of the three-stage type, and was driven from a crank on an extension of the main crank shaft. The cylinder diameters were 510 mm. 445 mm. and 115 mm. respectively, with a common stroke of 350 mm. The three scavenging pumps, 650 mm.

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* From an article by R. Dreves in a recent issue of "Zeitschrift des Vereines Deutscher Igenieure."

diameter by 500 mm. stroke, were driven by rocking levers from the crossheads of the three working cylinders, while the fuel pump, 18 mm. diameter by 60 mm. stroke, was driven from the cam shaft through excentrics. The rocking levers of the first and third cylinders also drove two bilge pumps and two pumps for the piston cooling water.

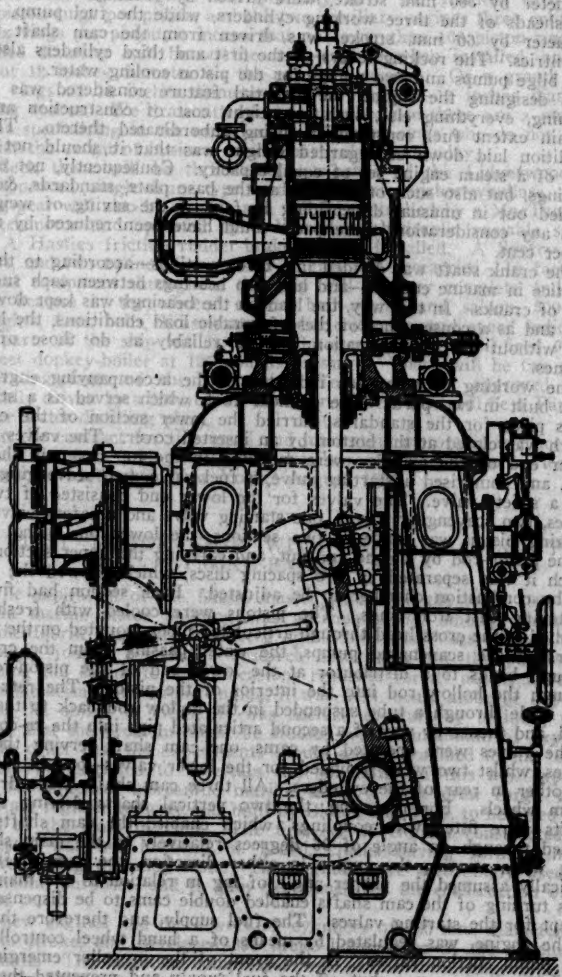
In designing the engine, the essential feature considered was reliable running, everything else, such as weight, cost of construction and to a certain extent fuel consumption, being subordinated thereto. The only condition laid down as regarded weight was that it should not exceed that of a steam engine set of equal capacity. Consequently, not only the bearings, but also such other parts as the base plate, standards, &c., were carried out in unusual dimensions; in fact, if the saving of weight had been any consideration, these parts could have been reduced by at least 50 per cent.

The crank shaft was divided into three sections—according to the usual practice in marine engines—and had two bearings between each successive pair of cranks. In this way, the load on the bearings was kept down very low, and as a consequence of these favorable load conditions, the bearings ran without forced lubrication just as reliably as do those of steam engines.

The working cylinders—as indicated in the accompanying engraving—were built in two parts. The cylinder foot, which served as a stiffening cross piece for the standards, carried the lower section of the cylinder, which was closed at the bottom by an inserted cover. The valves for the upper end of the cylinder were mounted in the top cover in the usual way, and comprised a starting valve, a fuel valve, two scavenging valves and a safety valve. The valves for the lower end consisted of two fuel valves, two scavenging valves, one starting valve and a safety valve. The working piston was made in two sections, the lower one being secured to the piston-rod by a cone and nut, and carrying the upper section, from which it was separated by two spacing discs, which allowed the volume of the combustion chamber to be adjusted. Each section had five self-tightening cast iron rings. The pistons were cooled with fresh water supplied to the cross head through articulated pipes mounted on the rocking levers of the scavenging pumps, the water passing from the crosshead through bends to a distributor at the lower end of the piston-rod, and through the hollow rod into the interior of the piston. The return flow was made through a tube suspended in the hollow rod back to the cross-head, and thence by way of a second articulated pipe into the re-cooler.

The valves were operated by cams, one cam shaft serving the upper valves, whilst two were provided for the lower valves—one in front and the other in rear of the cylinders. All three cam shafts were driven by worm wheels. For reversing, the two vertical shafts driving the cam shafts were fitted with couplings, which enabled the cam shafts to be turned through an angle of 36 degrees relatively to the main shaft, so that, when starting the engine in either direction, the cam shafts automatically assumed the proper angle of lag in relation to the main shaft. This turning of the cam shafts enabled double cams to be dispensed with, except for the starting valves. The fuel supply, and therefore the speed of the engine, was regulated by means of a hand wheel controlling the fuel pump suction valves. In the event of the propeller emerging in a rough sea, a governor shut off the fuel supply and prevented the engine from racing.

The engines were tested extensively, both in the shops and on board. Before erection in the ship, the first of the engines were put through a continuous six days' test, followed by a maneuvering run for twelve hours, reversing for three hours and four hours at slow speed. The



"The Engineer" Swan Sc

TWO-CYCLE DOUBLE-ACTING MARINE OIL ENGINE

fuel consumption amounted approximately to 8 ounces per brake horse-power hour, and the overall efficiency was 72 per cent. Taking into consideration the power absorbed by the scavenging pumps and the compressor, the mechanical efficiency was 89 per cent. For such comparatively small engines no better results could be expected.

In the tests on board ship, troubles and delays arose through the appearance of whirling oscillation to an extent unanticipated from previous experience. It was found impossible to run at certain speeds without endangering the whole equipment, unless these critical periods were traversed quickly. The conditions were aggravated by the circumstance that the full speed of 120 revolutions coincided with one such critical period, and in order to overcome this difficulty, considerable structural alterations had to be made. When these changes had been made, exhaustive measurements were taken with a Fraum torsion indicator with photographic attachment, an instrument which enables an accurate visible indication to be obtained of the phenomena occurring inside a shaft under the most difficult circumstances, such as when it is subjected to high-frequency oscillations or when the vessel is maneuvering ahead or astern.

In May, 1915, the *Fritz*, fitted with the engines was put through her first trial trip on the Elbe, the run lasting eight hours, and proving entirely satisfactory. The measured fuel consumption amounted to about 5½ ounces per indicated horsepower hour at a mean output of 1093 indicated horsepower for each engine. To test the maneuvering capacity, twenty-two starts and reversals were made in 33 minutes. Subsequent examination of the engines revealed the development of cracks in various parts of the cylinders. The cylinders were therefore renewed on lines suggested by the experience acquired by the builders.

By reason of the delays occasioned by the war, the second trial trip, with the re-built engines, was not carried out until April, 1919. This trip also went off satisfactorily, and the subsequent inspection showed the engines to be in excellent condition. Under the provisions of the Peace Treaty, the *Fritz* had to be surrendered to the Allies. The builders directed the attention of the British Government to the fact that the *Fritz* was an experimental vessel, and that, notwithstanding the previous tests, the reliability of the new type of engine could not be ascertained until after several sea voyages had been made. The suggestion was accordingly made that several trips should be made in the Baltic before surrendering the vessel. This suggestion was not accepted, and on November 8th, 1919, the ship left for England, where she arrived safely after encountering heavy weather. Since then she has been in commission for an English shipping firm. The builders of the engines and of the vessels are continuing their work, and have recently constructed large-cylinder units which have, so it is announced, satisfied the most stringent test conditions on prolonged runs.—"The Engineer," August 19, 1921.

CAVITATION.

Mr. E. R. Mumford, Superintendent of Denny Bros.' experimental tank, has recently been investigating the phenomenon of cavitation, which first attracted attention in the trials of H.M.S. *Daring* 30 years ago. On this 27-knot torpedo-boat destroyer Messrs. Thornycroft had fitted screws which absorbed sufficient power, according to their estimates, to drive the vessel at the desired speed, but failed by a knot and a half to do so. Mr. S. W. Barnaby, the famous naval achitect of the firm, felt satisfied that the bad performance was due to the formation of cavities in the

water forward of the screw, which cavities would probably be filled with air and water vapor. This view was confirmed by the very serious vibration of the stern when the engines were driven at full speed, although when the screws were removed the engines failed to shake the vessel when running at the same number of revolutions, showing that the excessive vibration was caused by some irregular action of the propellers. The cure was found in a great increase to the surface of the screws, the desired speed was easily obtained, and the excessive vibrations disappeared. Barnaby's theory is that the velocity with which water can be made to flow towards a submerged screw is due to the head of water over the screw, plus the atmospheric pressure, and there is, consequently, a definite limit to the speed to which it can attain. and a definite thrust per square inch of projected screw surface at which cavitation will commence. As the result of his experience, he considered that this limit of thrust was 11.25 pounds per square inch of projected area with the tip of the screw 1 foot below the surface of the water.

In his "Speed and Power of Ships," Admiral Taylor describes some experiments he carried out in the Washington tank in order to elucidate further this problem of cavitation. He arrived at the conclusion that tip velocity and shape of blade sections are the prime factors involved, and, further, that the cavities which caused the decided loss in thrust formed on the face of the blade and not on the back, as Barnaby had suggested. Mr. Barnaby truly points out that these experiments of Mr. Taylor's are largely discounted by the fact that it was not really a screw propeller that was used in these experiments, but really an instrument specially designed to produce cavitation. Further, he shows that the range of tip velocities in which cavitation may, or may not, occur is so large that it affords no criterion as to its probable appearance.

Professor Gümbel read a paper before the Institution of Naval Architects, in 1913, "On the Criterion for the Occurrence of Cavitation," in which he intended to show that its occurrence depended upon (1) the absolute pitch of the cross section of the blade; (2) the angle between the following edge and the face of the blade; (3) the amount of the slip speed as a function of the vessel's speed; (4) the temperature of the water and immersion of screw; and (5) *not upon the mean pressure per unit blade area or upon the tip speed.* In the discussion which followed Sir Philip Watts remarked that whilst thrust pressure is not, strictly speaking, a true measure of the cavitating speed, it nevertheless is a measure which can be readily used for comparing similar screws under similar conditions.

Mr. Mumford joined in the discussion and said that in actual practice pressures considerably higher than those suggested by Mr. Barnaby had been obtained before cavitation was observed. He suggested that when the speed of real slip was about 16 feet per second the phenomenon will manifest itself. His recent paper, read before the Institution of Civil Engineers, described some experiments carried out in the Dumbarton tank. Two model propellers, each about 4 inches in diameter and of 6 inches nominal pitch, the one having about twice the blade area of the other, were driven at such revolutions per minute in undisturbed water while not advancing as to draw air down from the surface. It was found that the smaller blades drew air from the surface when the nominal slip was 26.1 feet per second and the thrust per square inch of projected area 10.53 pounds, while the screw with double the area drew air from the surface when the nominal slip was 22.4 feet per second and the thrust per square inch of projected area was 5.14 pounds. The immediate drop in thrust when air drawing commenced amounted to 35 per cent and 40 per cent respectively. Another non-advancing model propeller, 3 inches

in diameter, about 4.3 inches nominal pitch, and projected area of .575, drew air in a vortex from the surface when the center of its horizontal shaft was immersed 2 feet, the nominal slip was 27.7 feet per second, and the pressure per square inch of projected area was 7.4 pounds. Accepting that these experiments afford some guide as to what might be expected in actual experience with full-sized propellers, it may be observed that the speed of nominal slip in the three cases is fairly constant, being respectively 26.1, 22.4 and 27.7. The pressures per unit of surface showed a much larger variation, being 10.53 pounds, 5.14 pounds, and 7.4 pounds per square inch. From these it appears that Mr. Mumford was justified in saying, in 1913, that the speed of the real slip gave a criterion of cavitation probably to be preferred to that of pressure per unit surface.

Mr. Mumford enunciates four kinds of cavitation, the first one being "when the relation of the size and immersion of the propeller to the power and thrust transmitted is such as to cause air to be drawn down from the surface—the cavities in such case being air-filled." This was evidently the kind of cavitation which was observed during these experiments. Mr. Barnaby pointed out that this class had really nothing to do with cavitation as he understood it, and that this phenomenon was quite distinct from cavitation and should not be confused with it. Mr. Mumford, in reply, said that he had come to the conclusion that a cavity was a cavity, whether filled with air from the surface or with air or other gases including water vapor from the water itself.

One thing is clear from these experiments, and also from those previously carried out by Admiral Taylor, and that is, that the phenomenon of cavitation can be produced in the tank. It is equally evident that although the experts differ as to the real cause of the occurrence, two cures can be had when it does take place. Mr. Barnaby has found that greater area of propeller blades gets over the difficulty, and Mr. Mumford that increase in diameter or blade area, so as to reduce the speed of the real slip of the ship's propeller to be under 16 feet per second, will avoid undue cavitation. Mr. Mumford is to be complimented on his interesting work and for his contributions to the discussions on this problem with which Mr. Barnaby's name has for long been associated.—"Shipbuilding and Shipping Record," September 8, 1921.

SHIPS' PROPELLERS: NOTE ON RESULTS OF EXPERIMENTAL RESEARCH REGARDING "CAVITATION" AS AFFECTING PROPULSIVE EFFICIENCY.

By Edwin Richard Mumford.

This note is confined to consideration of the causes and effects of "cavitation," as indicated by results of special experimental investigation. Models of ship, propellers, as usually tried in an experimental tank, give reliable results for comparison with those of non cavitating ships' propellers, by the application of Froude's well-known "Law of Comparison"; but when the conditions are such that the ship's propeller cavitates, the model screw does not cavitate at corresponding speeds and slips, because the atmospheric pressure on the surface of the water is the same for model as for ship, instead of being in the ratio of the dimensions; and as the degree of cavitation depends upon pressure difference per unit area, which is greater for ship than model (being in the ratio of the respective similar dimensions) the "Law of Comparison" is not applicable for direct comparison without correction.

It appears that there are four conceivable kinds of "cavitation," two of which cause the formation of persistent cavities in the water to the extent of reducing appreciably the mass of water acted upon in relation to the nominal slip of a propeller, and thus necessitates a relatively increased speed of slip, with consequent loss of propulsive efficiency in order to develop the desired thrust ($T = M V$). The other two kinds of "cavitation" are almost co-existent when they occur, and are probably the cause of the excessive and rapid erosion of the material of propeller blades that sometimes takes place.

The four kinds or conditions of "cavitation" may be described as follows:—

(*a*) When the relation of the size and immersion of the propeller to the power and thrust transmitted is such as to cause air to be drawn down from the surface—the cavities in such case being air filled.

(*b*) When the reduction of pressure immediately forward of the propeller is sufficient to cause air or other gases absorbed or diffused in the water to be freed and to concentrate in the vicinity of the propeller.

(*c*) When the reduction of pressure at the propeller is sufficient to cause momentary conversion of the adjacent water into vapor.

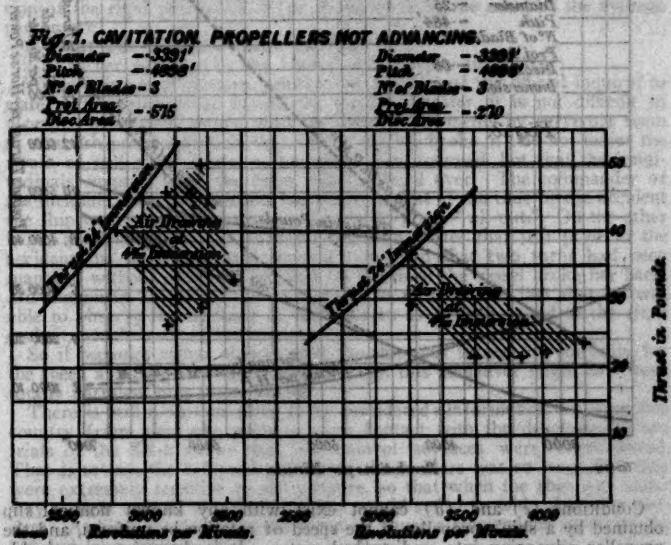
(*d*) When the reduction of pressure is sufficient to cause a nearly perfect vacuum.

In practice, condition (*a*) is a very common cause of loss of propulsive efficiency—probably much more common than is generally known—because, when it does happen it is not easily observed, owing to the turbulence of the water surface. Two model propellers, each about 4 inches in diameter and of 6 inches nominal pitch, the one having about twice the blade area of the other (projected area ratios 0.27 and 0.575 respectively) were driven at such revolutions per minute in undisturbed water while not advancing, as to draw air from the surface, the immersion of the center of the horizontal shaft in each case being 4.15 inches. It was found that the smaller blades drew air from the surface when the nominal slip was 26.1 feet per second, and the thrust per square inch of projected area 10.53 pounds, while the larger blades drew air from the surface when the nominal slip was 22.4 feet per second, and the thrust per square inch of projected area was 5.14 pounds. (See Fig. 1.) The immediate drop in thrust when air-drawing commenced, amounted to 35 per cent and 40 per cent, respectively, and the character of the thrust curve seems to be similar to that found by Admiral D. W. Taylor in the Washington Tank, and published by him in Figs. 220 to 224 in his book on "The Speed and Power of Ships." Another non-advancing model propeller, 3 inches in diameter, about 4.3 inches nominal pitch, and projected area ratio of 0.575 drew air in a vortex from the surface when the center of its horizontal shaft was immersed 2 feet, the nominal slip was 27.7 feet per second, and the pressure per square inch of projected area was 7.4 pounds.

Condition (*b*) frequently occurs with vessels of the fast cross-Channel type, torpedo-boat destroyers, and other vessels fitted with screw propellers impressing a high absolute speed of slip on the water acted upon. This phenomenon was made plainly visible by means of a stroboscope; air bubbles were seen to form at different parts of a model propeller of 3 inches diameter driven at high absolute slips up to about 40 feet per second nominal. The bubbles so formed were observed to persist as such in a spiral train in the race of the propeller.

In order to investigate this condition further, experiments were made with model screws with centers immersed 2 feet on a horizontal shaft in undisturbed water, and driven up to and beyond all known speeds of slip of ship's propeller, and at different speeds of advance. No air was drawn from the surface during these experiments. It was intended to measure the shaft horsepower and thrust in terms of the nominal slip of the driving

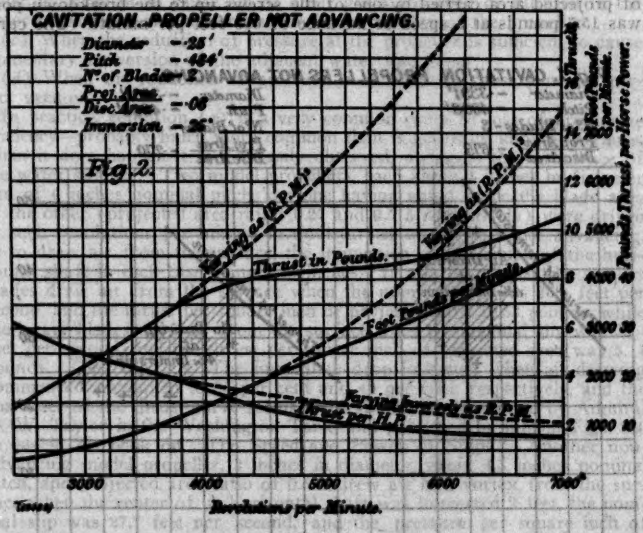
face of screws of different pitch ratios, but a difficulty occurred in connection with the measurement of the load friction of the apparatus, so that, for the present, only the curves of thrust are available. The results show a gradual reduction of relative thrust commencing at various nominal speeds of slip, and various pressures per unit area of projected surface, depending upon the ratios of pitch to diameter, of projected area to disc, and speed of advance. The commencement of the thrust breakdown occurs at higher nominal slip with increase of pitch ratio and increase of speed of advance. The maximum nominal speed of slip so far obtained at the commencement of breakdown of thrust is 34 feet per second associated with a pressure per square inch of projected area of 8½ pounds, which is the minimum so far obtained. The minimum nominal speed of slip obtained at the breakdown point is 17 feet per second associated with a pressure of 12.95 pounds per square inch of projected area and a speed of advance of 600 feet per minute. The maximum pressure per square inch of projected area carried by one of the screws up to the breakdown point was 15.7 pounds at a speed of advance of 600 feet per minute. The curve



of thrust beyond the breakdown point in all cases showed the falling-off to be gradual, and not in the least degree of the same character as when air is being drawn from the surface. Though the rate of falling-off in propulsive efficiency remains to be explored, it may be said that the one fairly reliable test shows that a 25 per cent increase of slip beyond the breakdown point involves a loss of about 20 per cent of propulsive efficiency, and the loss increases with further increase of slip. (See Fig. 2.)

Observations of undisturbed sea water subjected to reduction of atmospheric pressure under the glass bell of an air pump showed that bubbles began to form when the pressure was reduced by about 6 pounds per square inch, and that they increased in magnitude with further reduction

of pressure. This pressure difference at a propeller would give a speed of flow of water towards it of about 30 feet per second. In the turbulent wake of a ship the bubbles will probably be formed at a lower average speed of flow. The pressure of water vapor at 60 degrees F. is only ¼ pound per square inch. Assuming a propeller to be immersed 6 feet in water at such temperature, and impressing such a speed on the water as to create a region of absolute pressure of only ¼ pound per square inch, then the speed of flow of water towards that region due to the combined head of atmosphere and water, less the ¼ pound per square inch, will be about 50 feet per second, which is so much in excess of the average speed of slip of any ship's propeller that it appears probable that air bubbles extracted from the water by pressure difference, and thus reducing "M," are the usual cause of breakdown of thrust and efficiency of propellers of high-speed ships when not drawing air from the surface.



Conditions (*c*) and (*d*) cannot exist with any known nominal slip obtained by a ship's propeller if the speed of rotation be uniform, and the propeller advancing into undisturbed water. It is, however, conceivable and probable that minute cavities of extremely low internal pressure are created momentarily when impressing a high speed of slip in a turbulent wake. The alternating pressures during each revolution due to a varying speed of wake may set up vibration of the propeller blades which may further intensify the pressure differences. The minute cavities of low internal pressures thus created probably would not affect propulsive efficiency to an appreciable extent because they could not (like the air bubbles) persist as such, but would collapse almost as soon as created; the blows on the material of the propeller blades, caused by the collapse of such cavities, appear to furnish the only reasonable explanation of serious erosion that sometimes occurs (see paper by Sir Charles Parsons and Mr. Stanley S. Cook, in vol. lxi of Trans. Inst. N. A.).

It may be of interest to record that in order to make the cavitation tests with small models of 3 inches to 4 inches diameter, the full power of an electro-motor capable of giving 6 shaft horsepower was used.

Much remains to be investigated as to the effect of various factors—size, design, immersion and speed of advance—in relation to "cavitation" as affecting propulsive efficiency, but it is hoped that this note will prove adequate to evoke a good discussion of the subject.—"Engineering," July 8, 1921.

THE LESSON OF THE ZR-2 DISASTER.

Some Recent Facts Bearing on the Construction and Tests and the Conclusions Drawn Therefrom.

By LADISLAS D'ORCY.

The two chief questions which, in connection with the accidental destruction of the rigid airship ZR-2 (or R-38) come to the mind of the average person are:

1. What was the matter with the ZR-2?
2. Was her purchase by the United States Navy justified?

While it is yet premature, pending the findings of the official inquiry, to state with finality what caused the terrible disaster, it is not difficult to visualize what went wrong with the ZR-2. Statements by survivors seem to establish pretty definitely that the big airship broke in two not under the strain of full speed trials, as had first been assumed, but that the longitudinals snapped when the helm was put hard over. The commander of the dirigible, who was rescued, has declared that at the time of the accident the ship was making 50 knots—as against 65 knots "all out." On the other hand, another survivor, Lieutenant Bateman, states that just prior to the accident the ship had made turning tests, and that two turns had been managed without difficulty, but that on the third the vessel broke her back. His statement is particularly significant in view of the fact that he was able to observe the working of the rudders as he was seated in the stern cockpit, which is aft of the rudders.

So it becomes rather obvious that the ship was turned too suddenly for the speed at which she was flying, although this maneuver might have been totally harmless at a lesser speed.

There is still a further point to be considered: information reaching this country from men who were in close contact with the development and trials of the ZR-2 shows that her control surfaces were overbalanced. That is to say, the balanced portion of the rudders was so large that they were extremely sensitive to air pressure, so that when the ship was under way a slight turn of the steering wheel would suffice to whip about the rudders. At high speed this would naturally cause a tremendous strain on the 700-feet-long framework.

It would therefore seem that the design of the ZR-2's rudders was faulty. This defect, which might have been easily remedied, would not have been a serious matter on a stronger ship. But the ZR-2, far from being a strong ship, was what one may call an "extra-light" vessel—a feature which borders on structural weakness. To understand the why and wherefore of the situation we must look at the history of rigid airship development, which takes us back to Count Zeppelin.

Germany launched her first "super-Zeppelin," a 2,000,000 cubic foot vessel, in 1916, after fifteen years' experience in this branch of engineering, in which period she built some *sixty* Zeppelins ranging all the way from 400,000 cubic feet to 1,250,000 cubic feet. In the fall of 1916 one such super-Zeppelin, the L-33, was brought down fairly intact in England and

the British Admiralty instructed its airship designers to duplicate it. Up to that time British experience in rigid airship design and construction was limited to that obtained from a number of experimental ships that were being built after very incomplete drawings of pre-war Zeppelins. The British copy of the L-33, called the R-33, was only finished after the armistice, her trials taking place in the spring of 1919. Although the vessel embodied some improvements found in another captured Zeppelin, the L-49, which had come down intact in France, it should be pointed out that while the latter ship had a useful load of 39 tons, and the L-33 one of 30 tons, the British copy of these 2,000,000 cubic feet dirigibles had only a useful load of 24 tons. All of which is merely mentioned to show that a painstaking copy of an engineering structure will not necessarily be identical in all respects with the prototype—although they may *look alike*.

But while the R-33 carried a smaller useful load than her German sisterships, she seemed to be in every way as strong as the latter. How strong the hull of these ships was, the R-34 (sistership of the R-33) demonstrated at Mineola, Long Island, where for four days she withstood buffeting by winds, although on one occasion the anchorage fitting of the main handling rope was pulled clean out of the framework.

The success of the R-33 class airships prompted the British Admiralty in 1918 to prepare drawings for a much larger class of airships which were to be superior to the German L-71 type. This was the ill-fated R-38 (the ZR-2, as we call it) class, which incorporated numerous novel and original ideas. Now, it should be emphasized that when this class was *laid down*, all the experience the British had in rigid airship construction had been derived from copying German ships. The only firm which eventually was to produce a highly successful original design (Vickers, Ltd., with their R-80) had not by then emerged from the experimental period of their work; their experience was therefore unavailable.

And what may strike the reader as particularly odd, the Admiralty gave the contract for the construction of the R-38 (or ZR-2) to Short Bros., of Bedford—*a firm that had never before built a Zeppelin type airship* and whose entire experience with rigids was obtained from the building of R-31 and R-32, which were patterned after the *plywood-framed* Schuette-Lanz type.

Here then we have, in part at least, the answer to the question which heads this article: What was the matter with the ZR-2? *The ship was built in a factory that had no previous experience with duralumin airship construction, and to plans which were not based on practical experience.* To cite but one instance, the well-proven radial truss of the transverse frames was replaced on the ZR-2 by a tangential truss system, the merit of which had yet to be demonstrated. In this connection I cannot do less but pay a respectful homage to the memory of the late Colonel Campbell, chief airship designer of the Admiralty, who had sufficient faith in his ideas to go up on the ZR-2 during her several trial trips and who lost his life with the ship.

Knowing the circumstances which surrounded the construction of the ZR-2, we begin to understand why, as one report has it "several girders were strained in the factory when as many as thirty fitters crowded on them in the course of assembly work." It is quite conceivable that workmen accustomed to the resiliency of plywood girders would do just such a thing and that their foremen, not knowing any better, would not warn them. And a 700-foot airship is such a gigantic structure that the engineers familiar with the vagaries of duralumin—whom the Admiralty had detailed to the Short Works—could not personally supervise every detail.

For the sake of completeness it may be added that when Messrs. Short Bros. closed down their airship department, the Admiralty took over their factory and completed the ZR-2, whereupon she was handed over to the British Air Ministry.

That the hull of the ZR-2 was structurally weak was first demonstrated on the inflation of the ship when, due to unequal load distribution, several girders buckled. The failing members were repaired, but during the first trial flight trouble was again experienced from several intermediate longitudinals and transverse frames, so that it became necessary to reinforce certain portions of the framework. Details are not available as to the exact nature of this stiffening work, but one might suggest that by reinforcing certain girders others may have been further weakened. Of course, this is merely a guess.

Judging, however, from all that has been said before it appears beyond a doubt that the ZR-2 was structurally weak—a condition brought about by the desire to carry the greatest possible useful load. This, as originally designed, was to be in excess of 50 tons, but it was subsequently reduced by the fitting of a bow mooring gear, not to speak of the reinforcement of the hull.

As to the second question we have placed at the head of this article: "Was the purchase of the R-38 by the United States Navy justified?"—it would seem to the impartial observer that it was not.

Indeed, why should the Government spend abroad $2,000,000 on a foreign-built, untried type of dirigible?

On the one hand the Navy is desirous of developing rigid airships in this country. This can be brought about only through experimentation, and it will be admitted that if the necessarily heavy financial outlay has to be faced it will better serve its purpose if the money is spent here rather than abroad. American inventive genius is second to none in the world and can be relied upon to solve the problems of rigid airship construction just as well as it has solved other engineering problems.

On the other hand, if the Navy Department—which is in charge of rigid airship development to the exclusion of the Army—wanted to have a ready-made airship of proven design, it would seem that it could have secured from Germany, without cost, by virtue of America's participation in the victory—a dirigible that would have been far superior to the R-38. This will be seen from the appended table which gives the chief characteristics of the R-38 and of the L-71, Germany's largest Zeppelin, which was surrendered to Great Britain, while her sistership, the L-72, was surrendered to France:

Type	Capacity cu. ft.	Length ft.	Dia. ft.	Total H.P.	Useful Load (tons)	High Speed Miles per Hour
R-38	2,720,000	695	85	2,100	50(?)	75
L-71	2,420,000	745	79	1,740	48	75

It is not generally known that while the war spoils of the United States include a great number of airplanes and engines, the lighter-than-air material of Germany was entirely divided up between Great Britain, France, Italy and Japan, the United States merely playing the role of a disinterested spectator. That this was a grievous mistake, will be readily conceded by all those concerned with the development of American airships. —"Scientific American," September 17, 1921.

DIRECTION-FINDING WIRELESS.

By J. J. Bennett.

Direction-finding wireless, which came into use for naval purposes during the war, is now being applied to the service of the mercantile marine. The system has been adopted by France, America and Canada, each of which has a number of stations in operation. So far as the United

Kingdom is concerned, the Admiralty intend to maintain their direction-finding stations at the Lizard and at Carnsore Point. The stations at Flamborough and Berwick will also be retained until the end of the year. After that their continuance is uncertain, unless some other authority takes them over. As a fee of only 5s. is charged for giving a ship a bearing from one of these stations and the great value of them as an adjunct to navigation has been proved on many occasions, it would be regrettable if any of those esablished on our coasts had to be dismantled because nobody would undertake their upkeep. At the moment it seems probable that this is what will happen.

The principal use of direction-finding wireless as an aid to navigation is in enabling the bearing of a vessel in open waters, or when approaching pilotage waters, to be determined from one or more fixed points. When the bearings from two or more fixed points are obtained simultaneously the position of the vessel can be located by the intersection of the bearings. All bearings thus obtained by wireless telegraphy are the Great Circle bearings at the place where the bearings are observed. This means that in some cases, which will be explained later, the vessel will require special charts or special tables of correction. Independent of technical systems, there are three types of direction-finding wireless which may be employed as an aid to the navigation of ships. These are: (1) The direction-finding station type, in which a vessel transmits by her ordinary wireless installation to a fixed station, or stations, on shore that are fitted with directional wireless apparatus. The shore station determines the bearing and transmits it to a ship in the ordinary manner. Two or more stations may be grouped to determine simultaneously the bearing of the vessel from each station, and the intersection of the respective bearings, as transmitted by ordinary wireless telegraphy to the vessel by the master station of the shore group, enables the position of the vessel to be ascertained. In this case the responsibility for the accuracy rests with the station, and the vessel herself requires no special direction-finding apparatus.

(2) The beacon station type: In this the vessel is fitted with direction-finding apparatus by means of which the relative bearing of one or more ordinary fixed transmitting stations can be determined and the vessel's position located by the intersection of the bearings. These are necessarily obtained relatively to the ship's course at the time and are corrected accordingly. Any known ordinary wireless telegraphy station may be used for this purpose, or, preferably, certain stations known as beacon stations may be directed to transmit simultaneously, or successively, pre-arranged signals on certain wave-lengths at definite times during each hour. Under this system the responsibility for the accuracy of the bearing rests with the vessel, which must have a special direction-finding apparatus on board.

(3) The directional transmitting station type: In this method a rotating directional wireless beam having a fixed angular velocity is transmitted by a specially-fitted fixed transmitting station. The rotating beam, which swings round in similar fashion to the revolving beam of a lighthouse, has a sharply-defined zero position which passes through north and south at given times. Knowing the angular velocity of the beam, and by observing the time interval between the given times at which the zero passes through north and south and the time at which the zero signals are received in the ship, the bearing of the station can be determined. In order to ensure that the watches in the transmitting station and the receiving ships are synchronized the station transmits a timing signal first before commencing the rotating beam. In order to avoid calculation in the ship, and to obtain greater accuracy, a special watch, the face of which is marked in degrees, is required in the ship, and the scale corresponding to the angular velocity of the revolving beam. If this watch is

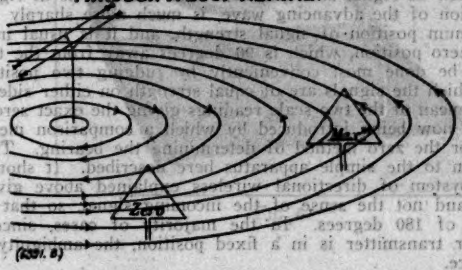
started at the moment indicated by the timing signal the bearing in degrees of the ship from the station can be noted from the watch at the moment when the zero signals are received. This bearing can be checked with subsequent zeroes. In this method the responsibility for the accuracy of the bearing rests with the ship.

The principles of direction-finding wireless and the technical methods of operating it may be epitomized as follows: Signals from an ordinary wireless transmitting aerial are propagated equally in all directions in a horizontal plane. The field of magnetic lines surrounding the aerial may be regarded as a series of concentric rings spreading outward uniformly from the transmitter; and the receptive powers of an ordinary vertical earthed aerial, apart from local or special conditions which may cause screening, may be regarded as equal in all directions. We will suppose that a receiving ship or station is fitted with a single loop aerial, i.e., a triangle as shown in Fig. 1, and that this aerial is not earthed, but that

Fig. 1. MAXIMUM CURRENT IN LOOP.

Fig. 2. ZERO CURRENT IN LOOP.

Fig. 3. FIELD OF MAGNETIC LINES OF FORCE THROUGH A LOOP AERIAL.



the two ends are connected to an ordinary receiving circuit so as to form a closed loop. A and B represent the two sides of this single loop and the arrows the direction of advancing wireless waves. If the plane of this loop be parallel to the direction of the advancing waves, the direction of the current set up in A will be in the same direction as that set up in B, so that the currents will tend to oppose each other. But if the two sides of the loop are spaced a certain distance apart the current induced in one side will reach its maximum an instant later than that in the other side and a resultant current will flow round the loop. If the loop now be turned through an angle of 90 degrees so that its plane is at right angles to the direction of the advancing waves which are shown

as facing or end-on arrows in Fig. 2, then it is clear that the electromotive force induced in the two sides will at any instant be equal and will oppose each other, so that no current will flow round the loop.

If the loop be turned through any other angle than a right angle then a resultant current will flow round the loop having a value which varies as the cosine of the angle which the plane of the loop makes with the direction of the advancing wave. Or, in other words, the loop aerial may be regarded as normally inductive to the field of magnetic force of the advancing waves, but as non-inductive to that field in certain positions. This is shown in Fig. 3, where for the sake of simplicity in drawing it is assumed that instead of being rotated the loop is moved round the transmitting station so that its plane is pointing at the transmitting station at the right hand, or maximum current position, and is facing the transmitter at the lower, or zero, current position.

Therefore, it follows that by means of a revolving loop aerial the direction of a distant station may be determined by the relative strength of the signals in different positions, and the single-loop aerial connected to receiving instruments and rotated round its vertical axis through a horizontal scale is the simplest form of direction-finding apparatus.

To increase the current through the loop it is usual to tune the loop with a condenser to the wave-lengths required to be received, and instead of a single loop a frame fitted with a multi-turn loop may be used. The larger the area of the loop the greater will be the resultant current induced. The revolving loop aerial system of direction finding has the advantage of simplicity and portability, but it does not lend itself so readily to the accurate reading of directions, especially when the loop is large and in consequence cumbersome. This system is not very suitable for work on board ship because the revolving loop requires to be set up in such a position that it is not influenced unequally during its orientation by other conductors in the vicinity. Here it should be noted that the zero position of signal strength, *i.e.*, when the loop is at right angles to the direction of the advancing wave, is much more sharply defined than the maximum position of signal strength, and it is usual in practice to read the zero position, which is 90 degrees away from the true bearing. This can be done most conveniently by judging two positions on the scale at which the signals are of equal strength on either side of the zero point, the mean of the two scale readings giving the exact zero. Improvements are now being introduced by which a comparison method is substituted for the zero method of determining the bearing. These involve an addition to the simple apparatus here described. It should be noted that the system of directional wireless explained above gives only the direction and not the sense of the incoming signal, so that there is an ambiguity of 180 degrees. In the majority of cases, since either the receiver or transmitter is in a fixed position, the ambiguity is of little consequence.

In the Bellini-Tosi system, in place of a rotary loop aerial two fixed loop aerials are employed which are connected to an instrument known as the radiogoniometer, or direction range-finding transformer. Inside the latter instrument is a small receiving coil attached to a pointer moving over a scale by which the direction of signals can be determined.

Let us take the case of a receiving set having two fixed loop aerials at right angles to each other. When the direction of the transmitting station is in the plane of one of the aerials maximum signals will be received on this aerial and zero signals on the other. Thus, if the signals are from a distant station at X or X_1 (Fig. 4), maximum signals will be received in coil A B and zero in coil C D. If the signals are from Y to Y_1, maximum signals will be received in coil C D and zero in coil

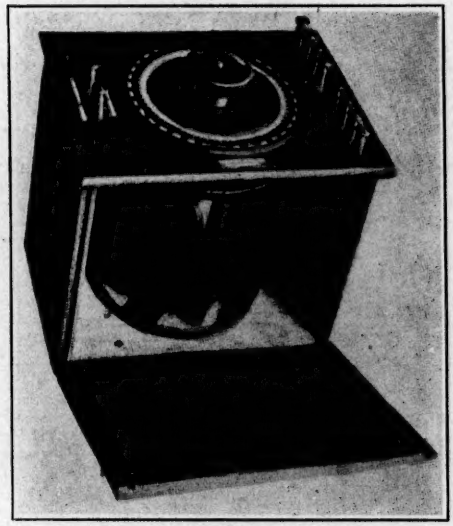
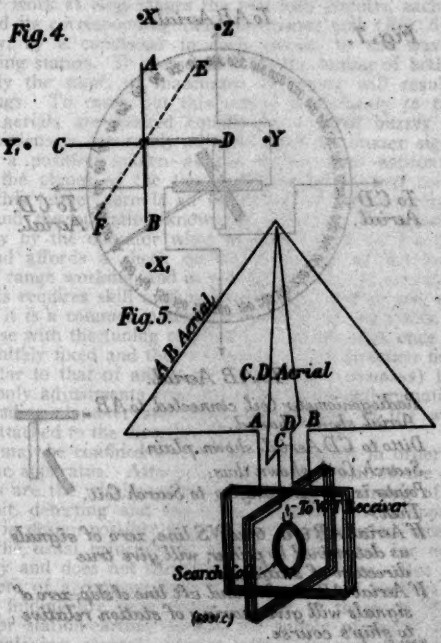


FIG. 6.

A B. When the direction of the transmitting station is not in one of these two planes, *e.g.*, at Z in Fig. 4, the signals will be received to a certain extent on both aerials, that is, a current will be induced in both aerials, the magnitude in each of which will be proportional to the cosine of the respective angle which each makes with the direction of the transmitting station. The two aerials are joined respectively to two small

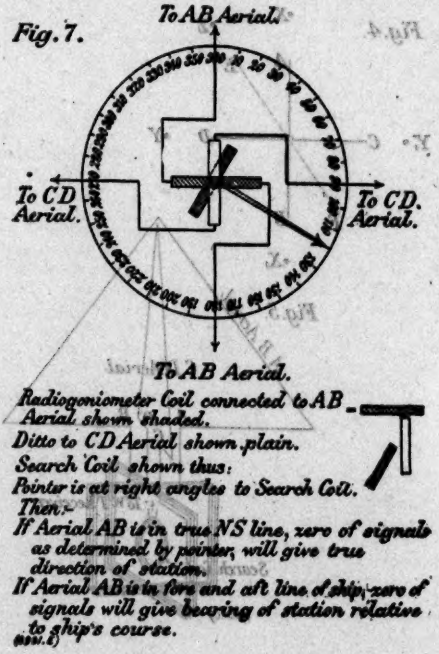


loops, each loop being a coil of several turns, fixed at right angles to each other in the radiogoniometer, so that this instrument forms a small replica of the large fixed aerials and reproduces faithfully the magnetic fields due to the currents induced in the two aerials.

Fig. 5 shows in diagrammatic form the radiogoniometer joined to two fixed triangular aerials, and Fig. 6 shows the actual radiogoniometer, or direction-finding box. The two fields due to the separate currents in the two fixed aerials combine to form a resultant field. In order that this resultant field shall have the true direction of the transmitting station, that is, in order that the resultant field due to the currents in the coils A B and C D from a transmitting station at Z (Fig. 4) shall lie along the dotted line E F, it is necessary that the two aerials should each receive their proportionate share of current. The receptive powers of both aerials must, therefore, be similar, and if the position of the direction finder is isolated from all other conductors, *e.g.*, set up ashore in a suitable

position, it is clear that this will be obtained by making the two aerials identical in all respects. If the aerials are set up in a ship the influence of other conductors in the vicinity of one aerial may be considerably greater than that affecting the other. Consequently, in order to keep the receptive powers of the aerials equal it may be necessary to employ a smaller loop for one aerial than for the other.

Fig. 7.


To AB Aerial.
To CD Aerial.
To CD. Aerial.
To AB Aerial.
Radiogoniometer Coil connected to AB Aerial shown shaded.
Ditto to CD Aerial shown plain.
Search Coil shown thus:
Pointer is at right angles to Search Coil.
Then:-
If Aerial AB is in true N S line, zero of signals as determined by pointer, will give true direction of station.
If Aerial AB is in fore and aft line of ship, zero of signals will give bearing of station relative to ship's course.

Assuming that the resultant field in the two aerials gives the direction of the transmitting station, then by joining the two coils of the radiogoniometer to their respective aerials so that the current in each aerial passes through its radiogoniometer coil, the resultant field can be faithfully reproduced in a concentrated form in this instrument. The direction of this field, that is, the bearing of the station, is obtained by revolving a small coil, known as the search or exploring coil, to which a pointer travelling over a scale marked in degrees, from 0 degree to 360 degrees, is attached. Fig. 7 shows the scale and diagrammatic plan of a radiogoniometer. It is usual to determine the zero position of the signals in taking directions, and if the search coil pointer is adjusted at 90 degrees to the plane of the search coil and if the aerials at the station are set up as shown in Fig. 7, the zero signal position as determined from the pointer will give the actual true bearing of the station or its reciprocal in degrees.

If the direction-finding apparatus is set up in a ship with the plane of the aerial in the fore and aft and athwartship lines the pointer will give the bearing of the station relative to the ship's course at the instant of receiving the signal, and the relative bearing must be corrected accordingly to give the true bearing.

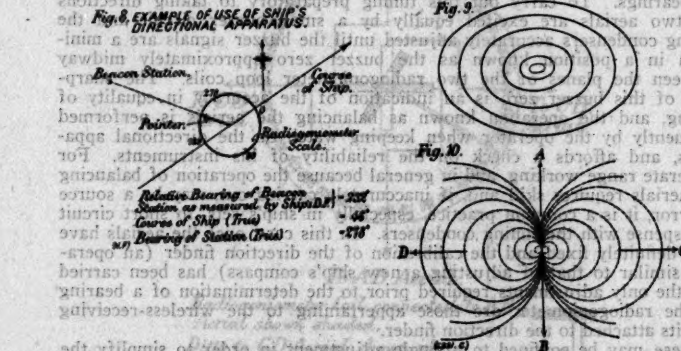
A means of ascertaining the course of the ship whilst taking the bearings must, therefore, be available. In order to obtain maximum sensitivity for work at long ranges the two loop circuits, each formed by one aerial and its corresponding radiogoniometer coil (Fig. 5) may be tuned separately, by a condenser in each circuit, to the wave-length of the transmitting station. It is essential that the tuning of both circuits should be exactly the same, as inaccuracy in tuning will result in inaccuracy of bearings. To carry out this tuning preparatory to taking directions the two aerials are excited equally by a small buzzer circuit and the tuning condensers accurately adjusted until the buzzer signals are a minimum in a position known as the buzzer zero approximately midway between the planes of the two radiogoniometer loop coils. The sharpness of this buzzer zero is an indication of the accuracy in equality of tuning, and the operation known as balancing the aerials is performed frequently by the operator when keeping watch on the directional apparatus, and affords a check on the reliability of his instruments. For moderate range working, and in general because the operation of balancing the aerials requires skill and, if inaccurately carried out, may be a source of error, it is a common practice, especially in ship's sets, to short circuit or dispense with the tuning condensers. In this case, once the aerials have been definitely fixed and the calibration of the direction finder (an operation similar to that of adjusting a new ship's compass) has been carried out, the only adjustments required prior to the determination of a bearing by the radiogoniometer are those appertaining to the wireless-receiving circuits attached to the direction finder.

These may be confined to a single adjustment in order to simplify the use of the apparatus. Attached to the search coil of the direction-finding apparatus are the wireless receiving circuits, consisting of a tuned receiving circuit, detecting and amplifying circuits. This apparatus, although differing in design possibly from the ordinary wireless-receiving apparatus, follows the usual principle of receivers employed for ordinary wireless telegraphy and does not therefore need describing. But since the receptive powers of a comparatively small loop aerial such as can be used in direction finding are very much inferior to those of the ordinary type of ship or station-earthed aerials, the supply of signal amplifying apparatus employing several vacuum valves is not only desirable but an essential feature of the direction-finding receiver.

The foregoing explanation covers the principles and operational methods of the types of direction-finding wireless telegraphy known as the direction-finding station type and the Beacon station type. Up to the present these types, which comprise directional-receiving aerials and apparatus either on shore or in a vessel, are the most commonly used. The procedure followed in communicating with a direction-finding type station is that the ship desirous of obtaining her bearing calls the shore station by her ordinary wireless telegraphy and asks for the bearing; the station indicates by wireless that it is ready to determine the bearing; the ship then transmits her call sign continuously for a short period, during which time the station ascertains the bearing by the use of its apparatus and transmits to the vessel by wireless her true bearing and the time at which the bearing was observed. If the ship requires simultaneous bearings from two directional shore stations so as to obtain her position, she calls up the controlling station of the group and intimates the fact. Both stations then determine simultaneously by their direction finders

the respective bearings of the ship. The controlling station collects both bearings (a land line is used between the shore stations) and either transmits to the ship the two bearings and the time at which they were determined, or, if equipped for the purpose, determines the position as obtained from the two bearings and transmits it to the ship.

With a station of the "beacon" type a ship fitted with directional-receiving apparatus can determine by the radiogoniometer of her apparatus, her bearing at any time convenient from a known ordinary wireless telegraphy station, or from a beacon station transmitting at routine times. The bearing as obtained is the angle between the course of the ship, *i.e.*, 360 degrees on the radiogoniometer or the fore and aft line of the ship,



and the direction of the station, and this angle must be applied to the true course in order to obtain the true bearing of the station (see Fig. 8). A similar procedure may be adopted with two or more stations when it is desired to obtain a fix, the bearing of each station being observed in turn.

In the place of either of these two types using an ordinary transmitter and a directional receiver, there is, as already mentioned, a third type of station using the converse of these two principles. Instead of the ordinary transmitting station from which signals are sent in all directions, a special station is designed capable of transmitting signals having a maximum amplitude in one direction and a minimum amplitude in a direction at right angles. Fig. 9 shows in plan approximately the magnetic field spreading out from an ordinary transmitting aerial of symmetrical form, and Fig. 10 that for directional transmitting stations of this type. A receiving station at C or D will receive the signals at maximum strength, and at A and B will receive the signals at minimum or zero strength. It may be convenient to regard this field as a broad beam of light having a maximum intensity at C D, and being obscured towards A and B. If, then, the field, or beam, shown in Fig. 10 is transmitted continuously while revolving at a fixed angular velocity with the zero, or obscuration, position passing through the line A B at definite times, a ship, knowing these times, will be able to find her bearing from the station by observing the time interval between the definite times mentioned and the time at which zero signals are received as the beam passes through north and south. For example, suppose the beam passes through the north-south line at successive exact minutes and that the watch in the ship has been exactly

synchronized with the time at the direction-finding transmitting station; then if the ship receives zero signals (that is, loses the sound of the transmitter in her ordinary wireless-receiving apparatus) at, say, each twentieth or fiftieth second during the first minute and succeeding minutes, then the bearing of the station is 20/60 or 50/60 of 360 degrees, i.e., 120 degrees or its reciprocal. By obtaining in the same manner her bearing from another directional transmitting station suitably placed and transmitting on a similar procedure immediately after the first station, the ship will be able to fix her position. The principle of the transmitting apparatus just described is somewhat similar to that of the Bellini-Tosi direction-finding receiver. Other principles may be employed, but the method of operation in the ship will be more or less similar. At the present time Great Britain has no directional transmitting station in actual operation. Germany had three of these stations during the latter part of the war.

The advantages of the directional-finding station type are that for using it a ship requires no special wireless telegraphy apparatus beyond her ordinary installation and no special skill or training is required by the ship's staff. The apparatus in the shore station is worked under stable and favorable conditions, the loops in the station can be made large and thus give greater accuracy and the station can carry a skilled staff practiced in taking bearings, thus taking responsibility and informing the ship. The disadvantages are that as ordinary wireless is used congestion on the wave lengths used by the stations may be caused when several ships are asking for positions, as any one set of directional-finding stations can only be used by one ship at a time. Further, ships can only use the stations at ranges well within the maximum of their transmitting installations. Also, if the shore stations signal the bearings only and not the position the ships will need, special charts or special tables of correction as the bearings obtained are the Great Circle bearings at the shore stations. Special charts, &c., will be unnecessary if the shore stations are organized so as to apply the corrections to the bearings and signal directly the position of the ship.

In the beacon station type the advantages are that any number of ships can obtain their bearing or fix their position simultaneously from one station or set of stations. A special wave-length can be allotted to the stations so that congestion with ordinary ship communications will not arise. Bearings and positions can be obtained at longer ranges, since the method is not limited by the ship's transmitting range and the stations may be made more powerful. The disadvantages of this type are that special directional receiving apparatus is required in a ship. This apparatus will need calibrating and occasional checking. Alterations in position of conducting material such as derricks, wire ropes, &c., in the immediate vicinity of the aerials may affect accuracy and necessitate re-calibration. The bearings are relative to the ship's head, and a ready method of obtaining the ship's course while actually determining the bearing must be available at her direction-finding installation. The ship's staff will need practice, and some technical skill in determining bearings, and must accept the responsibility for their accuracy. Ships must carry special charts or tables of special corrections as the bearings observed are those of the ship's end of the Great Circles.

The directional transmitting station type has the advantages that any number of ships can obtain their bearing or fix their positions simultaneously from one station or set of stations. Congestion of communications may be prevented by allotting the stations a special wave. Bearings and positions can be obtained at long ranges by making the stations sufficiently powerful. No special apparatus beyond a watch is required

in the ship. Although some practice and a little skill are required in taking the observations, they may be made directly by the bridge staff without relying on the ship's wireless operators. The disadvantages of the type are that a special kind of watch is necessary, and for accurate bearings exact timing of this watch and exact observation of the interval are required. Moreover, specially-constructed shore stations are required, and these must be suitably placed to enable a fix to be made. Ships will also need special charts or special tables of corrections as the bearings obtained are the great circle bearings at the shore stations.

As to the reliability of bearings obtained by the methods above described, errors may arise from two causes; first by mistakes in estimation of the bearing on the part of the operator of the direction-finding receiver; secondly, by wave distortion, *i.e.*, the direction of arrival of the waves at the receiver being distorted from the direct line joining the transmitting and receiving station. With regard to the first, practice and experience are essential. Assuming that the directional set has been calibrated and the apparatus and aerials are well maintained, the maximum error in fixed stations with practiced operators can be taken as 2 degrees, and the average error as less than 1 degree. In ships' sets with small aerials the average error is slightly greater. With respect to the second cause errors due to wave distortion are negligible by day when the bearings are over water. Bearings obtained during day and from a station suitably placed so that no considerable line of land intervenes can be accepted as reliable if they are competently taken. These are the best conditions for direction-finding wireless working. If choice permits, bearings should be obtained by day in preference to night, as night errors up to 5 degrees, and occasionally more, have been recorded. However, at short ranges of 50 miles to 100 miles over water the night errors are generally small, and bearings can be accepted as approximately correct. Times within 1 hour of sunet and sunrise should be avoided if possible. Bearings over land, unless it is flat, are less accurate than those over water. This is especially the case with night readings. High land close to and between the transmitter and receiver is a cause of error. Consequently, bearings should not be obtained when a vessel is close to high cliffs.

The estimated cost of a direction-finding receiving station is from 1,200*l.* to 1,500*l.*, and for a self-contained station with direction-finding receiver and transmitter complete from 3,000*l.* to 3,500*l.*, whilst that of a shore directional transmitting station available for long-range working and therefore requiring a powerful transmitting plant is approximately estimated at from 12,000*l.* to 15,000*l.* As regards ship's direction-finding sets, the apparatus required on board includes a twin direction-finding aerial system consisting either of fixed wires or of large rigid frames, a small room or cabinet (preferably not used for other purposes) and an operator. The complete receiving apparatus comprises wireless direction finder, tuning apparatus, receiving and amplifying set—these may be separate or combined; accumulator batteries for the receiver (4 volts to 6 volts, 50 ampère-hours to 100 ampère-hours, dry batteries, switches, spares, &c.; small charging and switchboard plant for accumulator batteries (these may be either charged from the electric mains of the ship or by small motor generator sets supplied from the ship's mains, or, if electric current is not available, a small internal-combustion generator will be required); and telephone or buzzer communications with the steering compass position, unless the receiving room is situated on the bridge with an immediate access to the steering compass. The cost of this apparatus with batteries, leading-in insulators and aerial is estimated at from 200*l.* to 300*l.* But this does not include cost of installing it.—"Engineering," September 2, 1921.

THE ZOELLY STEAM TURBINE CONDENSING LOCOMOTIVE.

An interesting locomotive is being put through trials on the Swiss Federal Railways, consisting of an engine fitted with a steam turbine and condensing plant. This engine has been designed and constructed by the Swiss Locomotive and Machine Works, Winterthur, in conjunction with Messrs. Escher Wyss and Co., of Zurich. Through the courtesy of the former firm we are able to give some notes on its de velopment.

The development is the outcome of work by Dr. Henri Zoelly, president of the Société des Ateliers de Constructions Mécaniques, Zurich, as a result of the coal shortage of 1918. A locomotive was placed, by the Swiss Federal Railways, at the disposal of the Swiss Locomotive and Machine Works, and Messrs. Escher Wyss and Co., for modification to Dr. Zoelly's designs. The locomotive frames carry the boiler in the usual way, and arranged in the front of the smoke-box, disposed across the machine will be noticed the steam turbine. The latter drives through double spur gearing a jackshaft placed across the frames, which in turn transmits the power to the driving wheels by means of side rods. The surface condenser is arranged between the frames under the boiler, the simplification of gear due to the use of the turbine making this space available. The condensing water, of which the temperature is raised by the steam, is cooled again by a current of air produced by the movement of the locomotive.

The apparatus in which this is effected is a long wrought iron casing, which will be seen on the tender, containing in its upper part a number of distributing tubes. Circulation is maintained by a small steam turbine-driven centrifugal pump which draws the water from the tender tank, passes it through the condenser and then passes it back to the re-cooler on the tender. The pipes in the latter have small perforations, and a fine spray of water is ejected into a current of air resulting from the speed of the locomotive. The loss of water as a result is said to be about one-half of the quantity required by the usual form of locomotive.

The locomotive has, up to the present, run trials aggregating about 471,667 ton-km., in ordinary railway service, and we are informed the condensing plant, circulating water re-cooler and the steam turbine have proved very successful. A good vacuum is said to have been obtained and maintained while running, and considerable economy over existing superheated steam locomotives is expected. Owing to the smaller consumption of water less has to be carried for the same distance, while the boiler has the advantage of being supplied with hot feed in the form of the condensate. This is at a temperature of about 50 degrees C., and is clean and free from oil. It is pumped to the boiler after having passed through a tubular feed heater, in which its temperature is further raised to 120 degrees C. As this condensate is used over and over again and the loss only has to be made good with fresh water, the incrustation in the boiler should be considerably reduced.—"Engineering," August 19, 1921.

IMPROVEMENTS IN DOUBLE BOTTOMS OF SHIPS.

Generally the strength of a ship's double bottom has to be greater to meet local conditions than would be required for it to take its part adequately in the main structure. Several attempts have been made for this reason to lighten the construction by reducing the cross-connection between the top, or inner, and the outer plating of the bottom. Accessibility has also to be kept in view, and with a view to improving this while providing ample cross-connection, Sir Westcott Abell, chief surveyor of Lloyd's Registry of Shipping, has devised the system of construction illustrated in the engraving. This system of construction consists in employing

several braced bracket frame floors between lightened plate floors. One of the latter is shown at the back of our figure, while four braced bracket frames are shown on the near side of the model, braced by means of longitudinal side girders, and rectangular bracket plates lightened by oval openings. In this design of double bottom framing a considerable part of the system of cross-connections is arranged in a longitudinal direction. The design enables the spaces adjacent to the intercostal girders to be kept free and thoroughly accessible. The longitudinal side girders can be fitted continuously for at least five frame spaces, and if double riveted at the ends a portion of them might be deemed continuous throughout, with a consequent saving of material. It is considered advisable at present to retain solid floors at every five frame spaces. One reason for this is that there is likely to be a line of hard points at the end of each rectangular bracket plate and also below the longitudinal girders, and the fitting of a through floor will localize the areas over which such rigid points might cause unevenness of stress distribution.

The design has advantages as regards construction. The bracket frames can be built in one piece on the ground and the intercostal girders slipped into place later. It also lends itself well to mass production and general templet work since the bracket plates can be made in numbers, similar for at least a good portion of the length of the hull. It is estimated that for a ship of 400 feet with only one side girder the weight of material would be little affected, but with a larger vessel, 500 feet, requiring two side girders, a probable saving of 50 tons might result with this construction.—"Engineering," July 8, 1921.

MECHANICAL REDUCTION GEARS FOR WARSHIPS AND MERCHANT SHIPS.*

By JOHN H. MACALPINE.

Since writing the foregoing paper valuable evidence confirming my whole position has been published. This is contained in Messrs. Walker and Cook's paper on "Mechanical Gears of Double Reduction for Merchant Ships" (vol. cxi, p. 369) read before the Institution of Naval Architects in March, and in the discussion of their paper (vol. cxi, p. 347); also in a letter by Mr. S. K. French, general superintendent, Propulsion Machinery Department, American International Shipbuilding Corporation, entitled "A Plea for Proper Setting of Double Reduction Marine Gears," (see "Mechanical Engineering," March, 1921, page 209).

Messrs. Walker and Cook's paper and its discussion give surprising evidence of how widespread the trouble with double reduction gears has been in Britain. Surely the authors exaggerate the gravity of the situation when they say that now "there are several double-reduction sets in actual service and running very satisfactorily" (vol. cxi, p. 369); and Admiral Goodwin remarks that "the experience of all the leading engineers in the country had been brought to bear on the problem" (vol. cxi, p. 347). The marine engineers of Britain have held the admiration of the world for a century, and my American friends will not misunderstand when I say I regard it as an honor and privilege to have been born and trained among them. Not a few I have known personally and many by reputation. I cannot believe that they have considered seriously a problem so simple and have failed to see the real difficulty—that is, the sensible yielding of the gear-casing under the changing forces to which it is subjected. From the first this seemed to me the plainly outstanding condition which had to be met in designing a good reduction gear.

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* Appendix to paper published in "Engineering," vol. cxi, page 609.

If in trying to find the important difference between single- and double-reduction gears the authors confined their attention to the three questions they specify (vol. cxi, p. 369)—viz. "(1) The energy of the turbines and gearing; (2) flexibility of the [propeller] shaft; (3) the end movement of the secondary pinions"—there need be no surprise that they failed (vol. cxi, p. 369). The elasticity of the casing, which has large influence on the action of all "rigid" gears, but more on double- than single-reduction, as is shown in my paper, is not considered by them. This elasticity is referred to only twice; once in the paper (vol. cxi, p. 370) and once by Sir Charles A. Parsons (vol. cxi, p. 348), and both times with commendation which is entirely undeserved. There is no compensation possible for poor tooth cutting, and consequent "irregularities in the teeth," or other imperfect machining, as is suggested; and, as I have pointed out in previous writings, if the casing changes configuration by the action of the gear it must be from altered bearing pressures which, in turn, follow from changed distribution of tooth pressure: hence, such yield as are commended, infallibly indicate a bad distribution of tooth pressure. It would be a total reversal of function if the tooth pressure were to keep the bearings of the rigid casing in line, as seems to be expected.

Captain Onyon, R. N., "had investigated many breakdowns, but had never found the fault to lie in the design" (vol. cxi, p. 348). This I can readily believe for one would expect to find good design; but, however good the design of a rigid gear, maintenance of close enough alignment cannot be assured. The authors emphasize by italics the great importance of correct alignment in the paper (vol. cxi, p. 369) and Mr. Walker in the discussion "particularly" wished to emphasize the fact that accuracy in cutting and aligning the gears was essential, and if this were attained all would be well" (vol. cxi, p. 348). No doubt the tooth cutting is of a high order and I am certain that in every case aligning has been most carefully attended to in the shop—and, in Admiralty gears, later in the ship, thereby recognizing its great liability to change. But, as the paper and discussion bear ample evidence, it is very far indeed from being all well. Our anticipation of this difficulty was based on simple theory, but I have shown in my paper that the facts were thereby fully accounted for. Mr. French's measurements, which I give in abstract below, have demonstrated the springing of the casing and consequent varying errors of alignment beyond the possibility of question.

[I may note in passing that the difficulty, so carefully considered by the authors (vol. cxi, p. 370) of the alignment of the intermediate shaft does not exist in floating frame gears. And that, if the reference to oil grooves (vol. cxi, p. 370) means that they cut these in parts of the bearing metal which take pressure, they thereby only allow the oil to escape from regions of higher to those of lower pressure, to the great detriment of the bearing action. The bearing should consist of two unbroken sheets, one in the top brass and one in the bottom, and the oil should be drawn in by the journal from two gutters, one on each side, where pressure never comes. Mr. Beauchamp Tower has shown, many years ago, that an oil sheet is then formed in the part under pressure and Professor Osborne Reynolds has demonstrated mathematically that it is stable within wide limits of pressure.]

I would gladly incorporate the whole of Mr. French's valuable letter if space permitted, as it deserves most careful study. His observations were made on the gear-case of S. S. *Carenco*, which was 12 feet 4 inches long and 8 feet 6 inches wide on the foundation. A very stiff tram, 14 feet long, was attached to the center of the casing cover, so that it could be placed first over one diagonal of the gear (starboard forward to port aft) and then over the other (port forward to starboard aft). Vertical movements of opposite corners of the casing relative to the tram were made by Starrett

indicators. The readings he gives are the means of the total deflections for five successive observations. In a table he records 19 observations taken on the voyage from Glasgow to Baltimore. These are for nearly all directions of wind and sea relative to the ship, and from no wind to a gale and no sea to rough; and in a heavy long swell with a light wind.

For no wind or sea all four readings were 0.002 inch, which Mr. French regards as showing the accuracy of the instrument. The character of the strain varied with the changing direction of the wind and the sea relative to the ship; apparently being principally dependent on that of the sea, as we would expect. "As the sea increases the deflectiveness of the gear-case increases in proportion." He also concludes that long swells "cause deflections fully as great as when the ship is operating in a storm." The largest deflections noted in the table, the rolling and pitching being heavy, were:—

Port forward, inch	0.035
Starboard forward, inch	0.022
Port aft, inch	0.035
Starboard aft, inch	0.021

There can be no question that these strains were caused by similar ones buckling the ship, longitudinally or transversely, or warping it—strains to which every ship is necessarily liable. Mr. French's method did not indicate the type of strain; but it, no doubt, partook of both buckling and warping. In the paragraph I quote below he is seen to be definitely of the opinion that the casing was warped. As I indicated in my paper (vol. cxi, p. 641), this strain is the one which most seriously affects alignment and to which the gear-case offers least resistance. Also, the gear troubles which are being discussed prove the prominent presence of warping which, indeed, is inherently probable. It would be instructive to measure in the shop the forces necessary to give a large casing a sensible strain of this type.

The following is a very instructive paragraph:

"While this ship was *en route* from Baltimore to Glasgow in loaded condition, very heavy following seas were encountered, in some cases running as high as 20 feet, at which time deflections of 0.050 inch were recorded. When gears of this type are so mounted that deflections of the kind under consideration occur, the case is so warped that the pinions are thrown at an angle with the driving face of the gear; and at a time when the overload becomes 100 per cent, due to surges of the sea on the propeller, and with about 50 per cent of the pinion face bearing on the gear face there is an overload on the driving face of the teeth of 400 per cent, this occurring approximately every 8 seconds."

If I am correct in the belief I stated (vol. cxi, p. 610) that 1/1000 inch is an important error of alignment of a bearing, even in a large gear, it will be realized how very far beyond allowable limits were the deflections noted by Mr. French. He is very conservative in his assumption that sometimes only half the tooth face was bearing. This is not the beneficent yielding of the gear casing "to ease off the effect of local stress" imagined by Sir Charles A. Parsons (vol. cxi, p. 348) but a very strong action due to the state of the sea.

It is not only by warping of the ship at sea that a rigid gear may be put out of line. The double check, in shop and ship, specified by the Admiralty, is proof of this. Slight heating of a bearing, unequal vertical or horizontal wear, the loosening or too tightly screwing up of a holding-down bolt, the grounding of the ship near the gear-casing, will all cause mis-alignment. All this I have pointed out in various papers, one of which was printed before the experimental gear was tried in 1909. And these are all of a kind which cannot imaginably be obviated by any method of setting the gear, as the title of Mr. French's letter may be thought to suggest.

When one considers all this—Mr. French's measurements and other causes of bad tooth bearing—there can be little surprise at the widespread trouble which has supervened, not at the low tooth pressures experience has proved to be necessary for rigid gears, of which the authors give examples (vol. cxi, p. 369); nor that experience has produced a call for lowering of these low values (vol. cxi, p. 369).

The *San Fernando* affords an excellent illustration of the foregoing. The impulses from the four-bladed propeller set up severe synchronous torsional vibration, and consequently after the first trial it was noted that the "gearing had shown signs of distress at four different points around the gear wheel" (vol. cxi, p. 347). Sir Charles A. Parsons pointed out the cause of the trouble, and the pitch of the propeller was increased from 18 feet 6 inches to 20 feet 6 inches, reducing the revolutions (vol. cxi, p. 369) so that synchronism was avoided. On the next voyage, instead of distress being shown at four points, the gearing was marked all round, and we are informed that this voyage was run in heavy weather (vol. cxi, p. 348). Sir Charles Parsons' explanation is that the shocks of the sea on the propeller blades now come at irregular intervals. But, there being now no synchronism, they would be very largely withstood by the inertia of the propeller and would reach the gear greatly mitigated. It seems to me incredible, considering the light tooth pressure of the *San Fernando* compared with that used in floating frame gears, that any detrimental marking would take place if unaccompanied by misalignment. Mr. French's observations leave little doubt that the true explanation lies in the warping of the ship due to the heavy weather of this voyage. Evidently the ship is sufficiently staunch and the tooth pressure small enough to keep within fair running condition in the average weather experienced, for we are assured that the teeth have polished up again (vol. cxi, p. 369).

How admirably Mr. Hadley F. Brown's evidence, quoted in my paper (vol. cxi, p. 641), dovetails into the foregoing.

The authors' remark (vol. cxi, p. 369) that the expression "pressure per lineal inch of tooth surface" (or, more correctly, tooth face) has no real value unless the pinion diameter is given. But its value can only be judged when the pitch and all other data of the teeth are given; and it may be especially misleading unless the ratio of length of tooth face to pinion diameter is stated. To take an extreme illustration: If we have a pinion of 10 inches diameter and tooth face of 6 inches we may obviously impose a high pressure per inch, as the torsional and deflection errors are quite negligible and, with good alignment and tooth cutting, the distribution will be excellent. But, if the tooth face is made 120 inches long the deflection and torsional errors will be very large and the tooth pressure under other than an extremely light load will be highly localized. For the same factor of safety the load per inch will be very different in these two cases. Still keeping the same factor of safety, the power transmitted by this pinion at a given speed will increase as the tooth face is lengthened from 6 inches; will reach a maximum; and, on still greater increase, will diminish. From careful estimates I concluded that the maximum is reached for floating frame gears and pinions with three bearings at a tooth face of about four pinion diameters—in this case about 40 inches (vol. ci, p. 457). (For a rigid gear this best ratio will be less and for a two-bearing pinion very much less.) Hence a small change in this ratio, lengthening or shortening the tooth face from what gives the above maximum, while it will not sensibly change the safe power transmitted or the power constant, will very sensibly change the safe tooth pressure per inch. Hence the power constant should be used in preference to tooth pressure; and, if the tooth face is short in any of

the gears referred to by the authors they were still more lightly loaded than is apparent from the constants given. I have discussed this question more fully in my Glasgow paper (T. I. E. and S. lxi, p. 345).*

Sir John Biles endorsed a plea made by Engineer-Captain W. Onyon (vol. cxi, p. 348) "for further information with regard to actual experience of high tooth pressures." Captain Onyon seems to doubt the authors' suggestion "that higher tooth loading was quite practicable." He asks—"Could the authors give him the names of any vessels with a tooth pressure constant [K of the empirical—D rule] of 220, which had run 20,000 miles at full power?" While not desiring to interfere with the authors' final reply I can say that many ships with floating frame gears have much more than fulfilled this condition.

I give in the subjoined table the data of 17 such ships:—

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* A good example of what I have stated is the Calderbank gear (vol. cxi, p. 369) which, apparently, the authors introduce to show that "considerably higher loads" (vol. cxi, p. 369) could be adopted than those at which some ship owners have baulked.

First, the Calderbank gear is a land gear, no doubt on a concrete foundation quite free from the warping which affects a ship's keelsons. Hence, even if it had been safe to judge from the pressure per inch and pressure constants given, 1,150 pounds and 298 pounds respectively, it would have had little bearing on the values allowable in a ship.

But only the pressure and diameter of the second reduction pinion are given, the speed, tooth face and number of pinion bearings, being omitted. The two latter are essential to a proper judgment.

The gear is of 750 horsepower and the total ratio of reduction is

$$\frac{2,000}{70} = 28.57 = 5.35^2$$

To keep the second-reduction gear small the second ratio might be kept as low as 4. It is improbable that it was over 5; but, if so, the pinion was still more lightly loaded.

Let l = length of tooth face in inches,
r = ratio of second reduction, and
C = the power constant.
Then $70\,r$ = R.P.M. of the 14.9-in. pinion.
We have

$$\frac{1,150\ l \times 14.9\ \pi \times 70\ r}{12} = 750 \times 33,000$$

or

$$l\,r = \frac{750 \times 33,000 \times 12}{1,150 \times 14.9 \times 70\ \pi} = 78.82$$

$$C = \frac{1,000 \times 750}{14.9^3 \times 70\ r} = \frac{3.239}{r}$$

If $r = 4$, $l = 19.7$, and C = 0.810.
If $r = 5$, $l = 15.8$, and C = 0.648.

Thus the tooth face is very short and this gear is under a very light load.

Reference to Fig. 2 of my paper will show that for C = 3.3, and $l = 4$ pinion diameters, the tooth pressure of a 14.9-inch pinion is 1,550 pounds per inch. With a power constant of 4, which has been quite successfully used, the tooth pressure rises to

$$\frac{1,550 \times 4}{3.3} = 1,879.$$

The horsepower is

$$P = \frac{14.9^3 \times 70\ r\ C}{1,000} = 231.56\ r\ C.$$

According as we choose C = 3.3 or 4 and $r = 4$ or 5, we get values of P from 3,057 to 4,631; that is, from four to six times the power of the Calderbank gear with the same diameter and speed of pinion.

Ship.	K for Second Reduction Pinion.	Ratio — Tooth Face / Pinion Pitch Diameter.	Sailed. Distance.	Sailed. Date of Report.	Ship Delivered.	Remarks.
Agawam	320	3·00	66,750	June, 1921	Dec., 1918	August, 1919, gears replaced.
Auburn	260	3·00	79,700	June, 1921	Jan. 1918	Gears in good condition. No trouble.
Avondale	260	3·00	80,200	June, 1921	June, 1918	No trouble with second reduction.
Cabegon	320	3·00	62,000	Nov., 1920	Mar., 1919	No trouble with second reduction.
Coakata	320	3·00	78,000	Sept., 1920	Dec., 1919	Gears in good condition. No trouble.
Cushnet	320	3·00	70,000	Apr., 1921	Dec., 1919	Gears in good condition. No trouble.
Fort Pitt Bridge	320	3·00	90,000	Feb., 1921	May, 1919	Gears in good condition. No trouble.
Hisco	260	4·01	120,000	June, 1921	Dec., 1917	No trouble with second reduction.
Inten	320	3·00	60,000	Feb., 1921	June, 1919	Gears in good condition. No trouble.
Ipswich	260	4·01	61,000	May, 1921	June, 1919	September, 1919, second reduction renewed. Poor material. No further trouble with second reduction.
Jefferson County	320	3·00	80,000	Jan., 1921	July, 1919	Gears in good condition. No trouble.
Laural	292	3·33	66,000	Sept., 1920	Mar., 1919	Gears in good condition. No trouble.
Mystic	260	4·01	70,000	Feb., 1921	Apr., 1919	Gears in good condition. No trouble.
Overbrook	260	4·01	107,000	June, 1921	Apr., 1918	No trouble with second reduction.
Sudbury	296	3·99	109,000	May, 1921	May, 1918	No trouble with second reduction.
Toledo Bridge	320	3·00	85,000	May, 1921	Jan., 1920	Gears in good condition. No trouble.
Wathena	320	3·00	65,000	Feb., 1921	Jan., 1919	August, 1919, gears replaced. Gears in good condition. No trouble.

Complete data of the ships with floating-frame gears would greatly extend this table, the large proportion of such ships having sailed over 20,000 miles. The mileage in many cases would have been greater, but it was found when collecting the data of mileage for this paper that the shortage of freight had caused a considerable number of the ships to be laid up.* No doubt the sailing distance to date of some of the ships is considerably greater than given in the table.

The values of K in the table are for the second-reduction pinions at contract power and speed and do not include the 10 per cent to 25 per cent reserve power in the boilers and turbines. Hence, in some cases, the actual values used may have considerably exceeded those given.

Under "Remarks" it will be found that the gears of the *Agawam* and *Wathena* were replaced soon after the ships were delivered. These gears originally had the long addendum teeth in the pinion which, for reasons given in the body of my paper (vol. cxi, p. 640), are very liable to score. The new gears were of the same design as the original except that the long addendum tooth was displaced by one placing the arc of contact in its proper position.

The Westinghouse engineers inform me that only in the case of the *Ipswich* did any teeth break in the second reduction pinions. In this case the steel had evidently been burned. Usually there is slight pitting which appears early and soon stops, after which the teeth continue in good condition with no sensible wear. In the few cases where there has been sensible wear it was found that the hobbing had not been of the highest class. Where lubrication failed the teeth usually scored but always rebrightened after it was re-established.

In studying what the authors say regarding vibration, one is impressed by the fact that the large part of the difficulty arises from the rigid connection of the turbine to the gearing, and from the great energy of rotation of the turbine. The propeller shaft is the most flexible part of the mechanism and they quite wrongly state (vol. cxi, p. 370) "to provide this flexibility in any other way is almost impracticable." As stated in the last paragraph of my paper, the simple introduction of an elastic shaft between the turbine and pinion it actuates completely avoids the authors' difficulty. It is a great safeguard against serious vibration: so much so that no evidence of synchronous vibration in marine floating-frame gears has come to my knowledge.†—"Engineering," July 29, 1921.

DURALUMIN: ITS PROPERTIES AND COMMERCIAL POSSIBILITIES.

By WILLIAM B. STOUT.‡

A material one-third the weight of cold-rolled steel yet with the same approximate strength characteristics; a metal which can be heat-treated to higher physical properties, yet by a process which does not take effect for an hour after treatment; a possibility for much lighter and at the same time cheaper production in certain lines of manufacture; and in a few words you have a picture of the future possibilities of the new copper-aluminum alloy known as "duralumin."

*The statement in the body of my paper that "many of the ships have sailed between one and two hundred thousand miles" (vol. cxi, p. 641) was founded on data furnished me in which the distance sailed had been in part estimated, not knowing of those laid up. I regret this exaggeration. It is found now that only nine ships are certainly known to have exceeded 100,000 miles.

†Only in the case of one geared turbine driving a generator on land, so far as I know, did the normal speed nearly coincide with a critical speed and produced considerable vibration. The alteration needed to change the critical speed was obvious.

‡Paper presented before American Society for Heat Treating, Indianapolis.

Receiving its first impetus in the development of aircraft abroad, and particularly by the Zeppelin firm, this metal has been perfected in this country to a point far ahead of the German product, while the work in aircraft of both the airship or lighter-than-air type, and aeroplane or heavier-than-air machine, has developed processes and methods now applicable to new lines of production.

The new alloy can be rolled into sheets or forged by hand press or power hammer. It can be cast, welded and soldered, while rivets of the same material used with it show new production possibilities on account of the peculiarity of the heat treat resultants. Connecting rods have been made of it, using the metal itself for wearing surface without anti-friction metal. These have operated successfully in both motor cars and aircraft engines. Worm gears have been used made of "dural" as it is colloquially called, and used in heavy truck service successfully against steel pinions. Timing gears of this metal show new possibilities of both wear and quietness, but the most use has been in structural shapes for ultra-light constructions. The metal itself costs about five times as much as cold rolled steel per pound. For production work, however, remember that but one-third the number of pounds is used for the same strength result, while the material being much easier to work than steel in most of its forms saves materially on labor, tool cost, and tool depreciation, and on enough production items so that in many cases constructions can be made cheaper of duralumin than of steel.

The work we have been following has been entirely along the line of minimum weight structures of fairly large size for aircraft work. The saving of a pound in an aeroplane structure means the addition of a pound of fuel in flight or a pound of pay load, so that in aircraft more than anywhere else minimum weight is a vital item. Strength is, however, no less fundamental, as an aeroplane in flight at high speed is subject to stresses far greater than those imposed on an automobile on the road or any form of present transport. Imagine having to build a motor truck of two-ton capacity that could run across a ploughed field at 70 miles an hour without breaking up. This alone would be some engineering problem, yet some modern planes of the bomb-carrying type are designed to do and do do this.

Structures are now being built of duralumin which far exceed former strength figures in wood and yet which are lighter than any previous wood-and-cloth aeroplane constructions. We have at present in process machines capable of carrying twenty passengers at two miles a minute for five hours, and fitted with six hundred horsepower engines, yet the whole machine weighs but the same as a Cadillac touring car. The entire secret of the weight result is of course not all in metal, but it is safe to state that the light weight figure could not have been reached with our present knowledge with any other known material.

Many have preferred steel in their experimental aircraft work on the basis that steel in tensile strength was stronger than dural even weight for weight. This is so, but the problem of strength in a structure is not ordinarily of tensile possibilities but of column or compressive strength.

For the same weight, duralumin has about three times the thickness of even cold-rolled boiler plate counterpart, and five or six times the section of alloy steels of high tensile. The rigidity of a sheet is dependent on its thickness very largely, and while dural is a much more flexible material than steel in equal sections, yet with the greatly increased sections used for equal strength a much greater rigidity is obtained than with steel.

For example, we have produced a rolled section, designed for a maximum column strength, and yet of such shape as to fit production requirements. This section of .035 thickness of metal in a 19-inch column length weighs 7¾ ounces and will support as a column 4 tons plus. It made of

steel of high alloy, its thickness would be, so far as tensile requirements go, about .009, a too great fragility to be trustworthy in a structure on account of the lack of rigidity of such a thin wall.

Duralumin, as we see it, looks very like aluminum except that it takes a high polish, and that when polished the glinting red of the copper in the alloy can be detected.

The polish in ordinary atmospheres is permanent, the metal being non-corrosive to a very high degree. In the tempered state it is almost impervious to salt spray, though in the annealed form salt water affects it.

The chief difference between this and previous aluminum alloys is its property of having its physical characteristics materially changed by proper heat-treatment. This heat-treatment in itself acts differently than with other metals, leading to some peculiar production possibilities.

Like all new things, this metal with its peculiar properties has certain assets and definite liabilities, but even its liabilities, in some cases, can be termed a profit, provided the structures made of the metal are designed to fit the peculiar requirements of the alloy.

The most marked difference in the metal is that the change in physical properties following heat-treatment is not instantaneous but very gradually rises to a maximum after about four days, and during this period not only is the tensile strength increased as high as 50 per cent or more, but the elongation increases from 600 per cent to 800 per cent. More than this, the metal in its heat-treated form can be re-heated and softened for passing through mechanical processes not of too violent a nature, and at the end of one hour come back to its original tempered characteristics. The fact that the extreme properties of the metal are not reached immediately, as in most metals, is of great advantage commercially, as will be explained later.

The heat-treatment of the metal, or tempering, as it might be called, consists of heating the metal to 920-940 degrees F. for from 7 to 30 minutes, the time being governed by the amount of metal in process. The material is then quenched in boiling water. It is then removed from the quenching bath and worked upon as soon as possible. In from one to two hours' time enough hardening will have taken place so that it will be difficult to work the metal. Where the time involved in making up the piece does not exceed one hour and where the bendings or hammerings are not too severe, this process is a thoroughly feasible production method.

This allows us in our aeroplane work, for example, to heat-treat a coil of sheet metal in a bath of sodium and potassium nitrates, quenching in an adjacent tank of boiling water and uncoiling the sheet or strip, start it immediately through the rolls which form it into the shape desired, having our complete process over and the spar ready for aging within 20 minutes from the beginning of the operation.

It is not my intention to go into any of the technical details or metallurgical study of the scientific side of this new alloy, but rather to point out the numerous possibilities opened up by its use and to describe and explain some of the results and processes now being obtained with duralumin, primarily in aircraft work. I will only state that the metal itself is nothing other than a copper-aluminum alloy, the copper running about 4 per cent, and some magnesium, but with zinc as the most detrimental component. For this reason, duralumin is made only from the purest 99 per cent aluminum, so that the impurities in each of the alloys from which the metal is formed will not build up a detrimental quantity. It is the inability of the Germans to obtain pure aluminum which has hindered their duralumin alloy from equaling the figures obtained in every-day production from the American product.

The metal itself, however, was developed in Germany from an original French laboratory development. The first real commercial use to which it

was put, so far as the public was concerned, was in the metal ships of the Zeppelin Company. One of the reasons for the slow development of the Zeppelin, aside from the problems of airship design itself, was the necessity of developing the metal along with the structure, and it is certain that the material, unless rightly handled, has as many idiosyncrasies as a New England spinster, as well as being as temperamental as a coloratura soprano from Grand Opera.

In the early metal considerable trouble was had with corrosion and with sheets, seemingly without reason, granulated, and until the properties and difficulties were worked out, structures built of the material were more or less uncertain and required frequent inspection. This is still true in very thin gauges which are not very carefully heat-treated, but with pieces of any real section, corrosion is now an almost unheard-of thing.

In the following outline I will give merely our own experiences with the present liabilities of the material and things to be watched out for in its use, but in criticizing I would, at the same time, state my belief that duralumin and derivatives which will spring from it will form undoubtedly the mainstay of all future light-weight developments in automotive and kindred structures, and that the use of this metal will grow by leaps and bounds as its marvellous advantages in both structure and production become very generally known and understood.

The main trouble had with duralumin is more or less similar to the problem met in other alloys, that is, the presence of impurities in the ingot, or air bubbles, small in their original forms, but which, when worked out and forged into thinner sections developed into serious defects.

Most of our material is of .035 stock, so that a very small speck of dirt can make a serious flaw in the metal sheet and a very small air bubble can make considerable pipe in the center of the sheet, almost impossible to find except by microscopic examination, added to considerable good luck.

Most of these flaws do not show up until after the rolling. In this case the greatest stress on the metal in putting it through the rolls is at a hidden point, so that to inspect these spars or chord members a dental mirror is used with a high light, and the surface very carefully examined. Frequently a hole as small as the point of a fine needle can be opened up into a flaw five inches long—a sort of stratification of the metal resulting from original ingot impurities.

At the beginning of our work, spar rejections, for this reason, ran as high as 400 per cent, this gradually being reduced, and it seems possible in the near future that our rejections will be normal. In all these sections there has been no attempt at drawing the metal, but merely bending it, as any attempt to draw the metal results in an extreme number of rejections through cracking. It is peculiar that these cracks may not appear until a number of hours after the piece is made, so that after rolling four days is allowed to elapse before parts can be inspected for cracks or flaws, or before any piece is allowed to be put into the actual aeroplane structure.

In the design of the rolls also great care must be taken against too great a degree of draw between annealings, and that all radii are of sufficient amplitude for the metal.

The great drawback of the material as we are using it at present is the presence of ingot flaws, rolled out in the sheet, which lead to a very high rejection cost. When this is cured, we can say, I believe, that the experimental stage of duralumin is over. As soon as quantities of the material are demanded also rolls can be put in for greater widths of sheet than at present available—sixteen inches being now the best obtainable—and tubing and other structural forms can be made available.

I believe that this material, and developments from it in better alloys, will eventually be more universally used than steel for structural requirements. Its use in aeronautics is already definitely established, new as its

use is, and the all-metal plane already given preference to the older types. I believe we will next see it in motor cars, at a saving of over half the weight, and in boats and busses, street and railway cars, bridges and girders. For the time being, however, it is a material to watch and study, and to make use of conservatively in conservative lines of engineering.

Composition:

Copper, per cent	3.50 to 4.00
Magnesium, per cent	.2 to .75
Manganese, per cent	.40 to 1.00
Aluminum (99 per cent pure)	Balance

Physical properties:

Specific Gravity	2.80
Weight, pounds per cubic inch	.102
Melting Range Centigrade	540 to 650
Compressive Strength tempered	44,000
Shear Value tempered	30,000
Tensile Strength tempered	50-60,000
Per cent elongation tempered, per cent	16 to 20
Modulus elasticity	10,600,000
Coef. expansion, per degree C	.00000226
Yield Pt.	30,000

—"Aerial Age Weekly," October 10, 1921.

CONTRIBUTION ON THE CALCULATION OF CRITICAL TORSIONAL NUMBER OF REVOLUTIONS.

By Fr. Sass, Engineer-in-Chief of the AEG Turbine Factory, Berlin.

The present paper presents a process for greatly shortening the time required for computing the critical torsional number of revolutions of multiple crank engines. The process is based upon the combination of any desired number of small masses into a single substitute mass of known magnitude which varies with the rate of oscillation. Tables are given for the rapid determination of the reduced length of the cranked portions of the shaft.

The investigation of power engines for critical torsional number of revolutions was first given special attention in Germany during the war when it was found necessary in building engines for submarines to carefully redesign their shafts for torque vibrations which always indicated dangerous rates of rotation. The data collected by the German oil engine industry in this connection is of value at the present time because oil engines are being more and more used in merchant ships, and an engineer who holds a responsible position in connection with the building of such engines is in duty bound to assure himself that his engine is not in danger of reaching a critical rate of rotation. Gumbel has furnished a graphic process which enables us to ascertain the characteristic rates of oscillation (Eigenschwingungszahlen) and also the critical rates of rotation of a shaft carrying any desired number of masses. Geiger has supplemented the process and among other things furnished data regarding the conversion of the cranked portions of the shaft into torsionally elastic plain shafts of equal power. Strictly speaking, this process can be regarded only as an approximation, since it ignores the damping forces and also makes certain assumptions, such as the symmetrical distribution of the separate masses about the axis of rotation which, in actual practice, is not

true of all masses. This does not, however, lessen its value because experience has shown that the critical rates of rotation computed in accordance with this process deviate but very slightly from those observed. In many of the experiments made by the Allegemeinen Elektricitats-Gesellschaft the difference between the computed and measured rates of rotation in the most important instances amounted to not more than 2 per cent. The simplifying assumptions are therefore fully justified, especially as without them it would have been practically impossible to make the calculations.

In spite of these simplifications, however, the process requires considerable time, especially in the case of multiple crank marine engines having several heavy masses attached to them, such as fly-wheel, dynamo armature, propeller, etc. In addition to the computation of the installation for the conditions to be met on the testing stand, perhaps with water brake, it will frequently be necessary to retest the installation thoroughly after it is built into the ship. Even though in such cases a large part of the data first obtained can sometimes be used in the second test, this cannot be done when the critical rates of rotation in the two cases show a marked difference, for the reason that in making the first computation the work will have been limited to the investigation of the nature of the oscillations in the vicinity of the characteristic rates of oscillation, so that the second arrangement, in which the characteristic rates of oscillation may perhaps be quite different, will have to be computed anew. Since the Gumbel process is a process by trial (the installation is designed for an assumed rate of oscillation and the completed engine is then tested to see if the conditions for a given characteristic rate of oscillation are fulfilled) it is sometimes necessary to repeat the computation several times for different rates of oscillation.

The following abridged process which has been used with advantage in the design of quite diverse power engines—four-stroke motors for submarines, two-stroke engines with pistons working in opposite directions, geared steam turbines—is based upon the combination of all the smaller adjacent masses, such as the compressor cranks and all the working cranks, or all the steam turbine discs, into a single substitute mass of known magnitude, which is conceived as acting at the place occupied by the outermost mass. The intersections of the R_n curve with the axis of abscissas are computed for this substitute mass instead of the seven to eight or more separate masses. Much time is thereby saved.

The substitute mass is different for each rate of oscillation and decreases as the rate of oscillation increases. This can easily be ascertained for a given installation by drawing the force polygons for from three to (at the most) five different, arbitrarily assumed rates of oscillation which will represent the nature of the oscillations, but only from the first mass (for example the compressor crank) to the first heavy mass (such as the fly-wheel).

Figure 1 shows the distribution of masses in a six-cylinder Diesel engine with a compressor crank and three appended heavy masses. Let the mass m_1 be the compressor crank, m_2 to m_7 the working cranks, m_8 to m_{11} the heavy masses. Let all masses be reduced to the radius r, the lengths to a uniform shaft diameter having the polar moment of inertia J_0. Let the amplitude of oscillation of m_1 be equal to $a_1 cm$ for all rates of oscillation. We next ascertain the amplitude of oscillation as a_2 for the mass m_2 by plotting in a diagram of forces, Fig. 2, for the polar height $H = \frac{J_0 G}{r^4}$ (in which G is the shearing strength "Schubkraft" of the shaft steel), the force

of inertia $T_1 = m_1 \omega^2 a_1$ and by drawing the parallel A_1A_2 at 01. ω is then the angular velocity corresponding to the rate of oscillation n. For $T_2 = m_2 \omega^2 a_2$ we have point 2 in the diagram of forces and with it as parallel, the section $A_1 A_2$ of the oscillation curve. We proceed in this manner and finally reach the point of intersection B of the oscillation curve with the location of the first heavy mass m_8. If we now connect the points A_1 and B by a straight line (broken line in Fig. 1) and in the diagram of forces

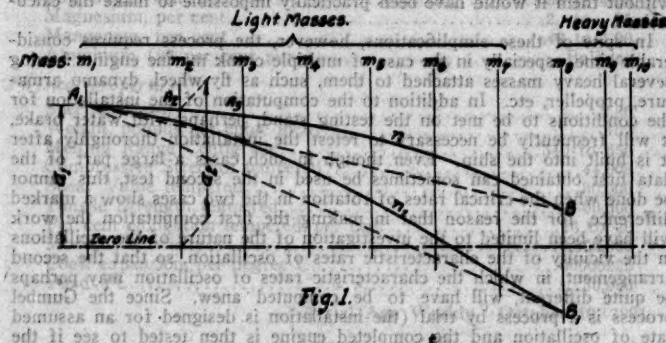

Fig. 1.

Fig. 2 Fig. 3.

draw OC parallel with A_1 B, we see that we can also find the point B directly if we plot a force of inertia $T' = CD$ in the diagram of forces. T' in this connection is always smaller than $\Sigma (T_1 \ldots . T_7)$ for the reason that the inclination which the straight line A_1 B makes with the zero line, at least for the rates of oscillation under consideration here, must always be less than the inclination of the last straight section of the broken oscillation curve A_1 B.

We now compute for certain rates of oscillations assumed to be at suitable distances from each other n, n_1, n_2, etc., (for example, $n = 1{,}000$, $n_1 = 3{,}000$, $n_2 = 5{,}000$, $n_3 = 7{,}000$), the oscillation curves A_2 B, A_1 B_1, etc., by the aid of the forces of inertia T_1 to T_7 up to the first large mass m_8, draw the straight lines A_1 B, A_2 B_1, etc., and in the diagram of forces the parallels to them OC, O_1 C_1, etc., Figs. 2 and 3. Forces of inertia T', T'$_1$, etc., are thus marked off which stand in a definite relation to $\Sigma (T_1 \ldots T_7)$. If we now take the quotients

$$X = \frac{\Sigma (T_1 \ldots . T_7)}{T'}$$

for the different rates of oscillation and plot these quotients above the rates of oscillation n, we obtain the curve $X = f(n)$, Fig. 4, which descends as the rate of oscillation increases.

It still remains to ascertain the substitute mass $m_1 \text{—}_1'$ acting at 1. It will be found from the general equation $T = m\,\omega^2\,a$; which for $m_1 \text{—}_1'$ reads: $T_1 \text{—}_1' = m_1 \text{—}_1'\,\omega^2\,a_1$.

ω^2 will be determined by the choice of n; $T_1 \text{—}_1'$ will be taken from the diagram of forces. We thus obtain—

$$m_1 \text{—}_1' = \frac{T_1 \text{—}_1'}{\omega^2\,a_1} \quad kg\ sk^2/cm.$$

If we plot $y = \dfrac{m_1 \text{—}_1'}{\Sigma\,(m_1 \ldots m_7)}$ above n, we obtain a second curve descending with increasing n, $y = \varphi\ (n)$, Fig. 4, by which the substitute mass $m_1 \text{—}_1'$ can be computed for each rate of rotation since $\Sigma\,(m_1 \ldots m_7)$ is known.

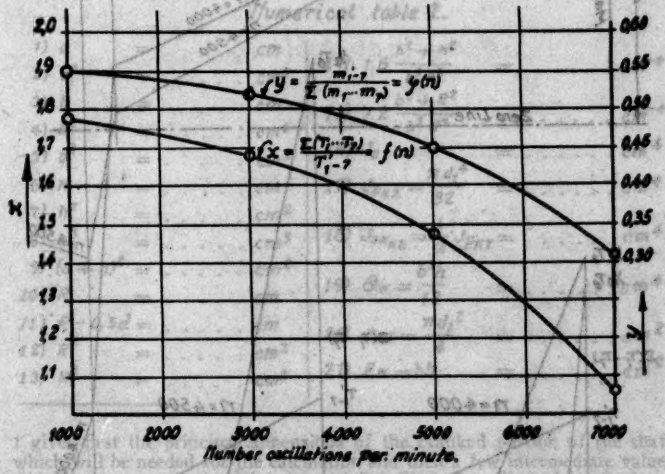

Fig. 4.

In computing the characteristic rates of rotation we first work with the masses $m_1 \text{—}_1'$, m_8, m_9 and m_{10}. The subsequent calculation follows the ordinary process. We merely substitute for m_1 the value $y\,\Sigma(m_1 \ldots m_7)$, in doing which y will be taken from the curve $y = \varphi\ (n)$ for the rate of oscillation in question and we get with $T_1 \text{—}_1' = y\,\Sigma\,(m_1 \ldots m_7)\,\omega^2\,a'$, also the direction $A_1 B$. The masses m_2 to m_7 do not appear in the subsequent calculation. In computing the direction of the section 8—9 of the oscillation curve, it is only necessary to see to it that the force of inertia $m_8\,\omega^2\,a_8$ is not applied at the terminal point of $T_1 \text{—}_1'$, but at the terminal point $\Sigma\,(T_1 \ldots T_7)$ which will be obtained from $\Sigma\,(T_1 \ldots T_7) = X\,T_1 \text{—}_1'$ by the aid of the curve $x = f\ (n)$. After reaching $T_{10} = m_{10}\,\omega^2\,a_{10}$ we automatically obtain the residual force R in its true magnitude.

Figures 5 to 7 show the graphic computation of the characteristic rate of oscillation of the first degree in a ship dynamo with opposed cylinders supplied by the A. E. G. for the steamer *Bismark* under construction by Blohm & Voss. It is surprising how few lines are required after ascertaining the curves x and y, to find the intersection of the R—n curve with the zero line. For $n = 4{,}000$ there is still a positive residual force; for

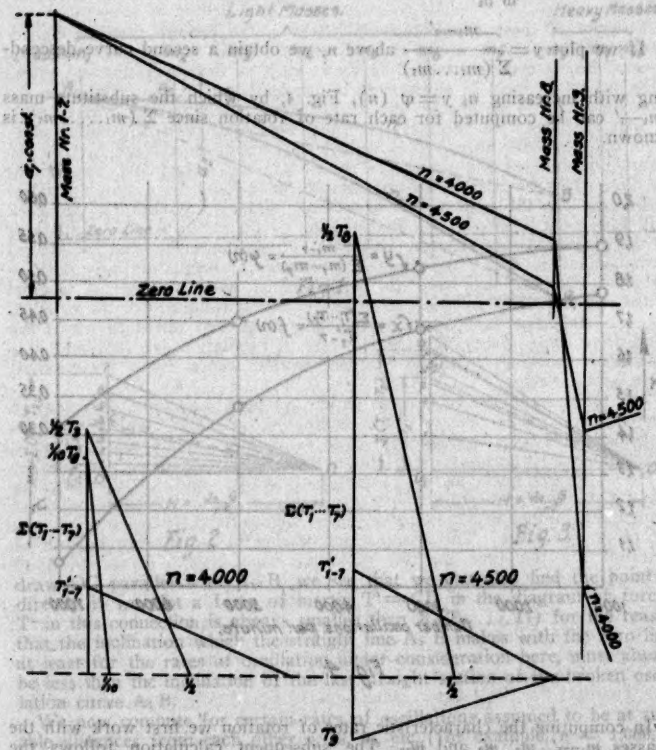

Fig. 5 to 7.

$n = 4{,}500$ it has already become negative as shown by the direction of the terminal tangent; the characteristic rate of oscillation of the first degree must lie between these two. It amounts to 4,440. The most dangerous critical rate of rotation of the first degree second order (since the engine has two cylinders) lies at 2,220 revolutions per minute. It lies so high that no resonance of any kind is to be feared throughout the entire rota-

tion field of the engine ($n = 400$). In fact no resonance was at any time indicated by the torsiograph and the running of the engine was entirely smooth and regular.

The above described process, the great accuracy of which can be attained only by a graphic process, will be found most useful in the calculation of the critical rates of rotation of all multi-crank internal combustion engines, but it can also be used, for example, in investigating the torsional oscillations of steam turbines with reduction gear. In such cases all the wheel discs of the turbine are combined in a single substitute mass acting on the first rotor and by means of this substitute mass the pinion gear wheel and other gyrating masses are computed.

Finally, an example will be given of a method of quickly ascertaining the equivalent length of the cranked portion of shafts, *i.e.*, the length of cranked portion which bears the same relation regarding torsional vibration as a plain shaft of circular cross section. This process—with a few modifications—is based upon the formulae furnished by Geiger. Table No.

Numerical table I.

1) a	= . . . . . cm	14) $1.6\frac{b^2+h^2}{b^3h^3}$	= . . . . . cm^{-4}
2) a^3	= . . . . . cm^3	15) $7.2\frac{b^2+R^2}{b^3R^3}$	= . . . . . cm^{-4}
3) b	= . . . . . cm	16) J_0	= . . . . . cm^4
4) b^2	= . . . . . cm^2	17) $J_{pKZ}=\frac{\pi d_1^4}{32}$	= . . . . . cm^4
5) b^3	= . . . . . cm^3	18) $J_{axKZ}=\frac{1}{2}J_{pKZ}$	= . . . . . cm^4
6) h	= . . . . . cm	19) $\Theta_W=\frac{b^3h}{12}$	= . . . . . cm^4
7) h^2	= . . . . . cm^2	20) $F_{KZ}=\frac{\pi d_1^2}{4}$	= . . . . . cm^2
8) h^3	= . . . . . cm^3	21) $F_W=bh$	= . . . . . cm^2
9) $(a+h)^2$	= . . . . . cm^2		
10) R	= . . . . . cm		
11) $R-0{,}5d$	= . . . . . cm		
12) R^2	= . . . . . cm^2		
13) R^3	= . . . . . cm^3		

1 gives first the principal dimensions of the cranked portion of the shaft which will be needed for the calculation and then a few intermediate values which will appear in the subsequent work. The values for a, b, h, R, d and d_1 correspond to the mass lines incorporated in Figs. 8 and 9. J_0 is the polar moment of inertia of the shaft diameter, to which Θ_W the moment of inertia of the crank webs (cheeks) are to be reduced. The index p indicates a polar moment of inertia and the index ax an axial one. The exponents KZ and W indicate that the quantity in question relates to the crank pin and crank webs, hence F_{KZ} relates to the cross section of the crank pin. If the latter is hollow account must be taken of this fact.

In Table No. 2 k is the quantity sought, whose development is portrayed in the table. Comparison with Geiger's deductions shows that k is used here in a different manner from what it is in Geiger's work. In our case k has the logical physical meaning of the lever arm by which the torque M conveyed through a doubly supported cranked portion must be divided in order to obtain the bearing pressure A. M and A do not occur mathematically in either the calculation of the reduced length of

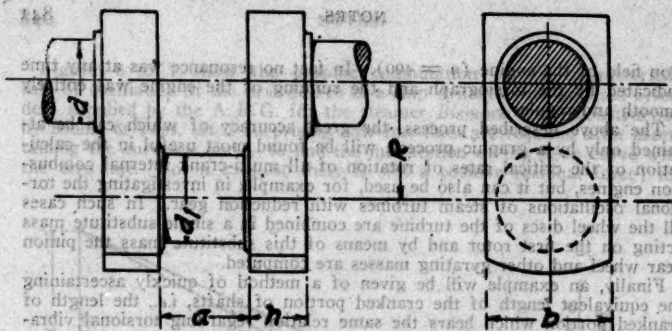

Fig. 8 and 9.

Numerical table II.

1)	$\frac{R^3}{7{,}8\ \Theta_W}$	= ..... cm^{-1}
2)	$\frac{1{,}2\ R}{F_W}$	= ..... "
3)	$1.8\ \frac{b^2+h^2}{b^3 h^3}\ (a+h)^2\ \frac{R}{2}$	= ..... "
4)	$\frac{a^3}{62{,}4\ J_{ax_{KZ}}}$	= ..... "
5)	$\frac{0{,}6\ a}{F_{KZ}}$	= ..... "
6)	$\frac{R^2 a}{2 J_{\rho KZ}}$	= ..... "
	$\Sigma(1 \ldots 6)$	= ..... cm^{-1}
7)	$\frac{R^2}{5.2\ \Theta_W}$	= ..... cm^{-2}
8)	$\frac{a\ R}{2 J_{\rho KZ}}$	= ..... "
	$\Sigma(7 \ldots 8)$	= ..... cm^{-2}
	$k = \frac{\Sigma(1 \ldots 6)}{\Sigma(7 \ldots 8)}$	= ..... cm

Numerical table III.

1) $2l_{red1} = 7{,}2\,\frac{b^2 + R^2}{b^3 R^3}\,hJ_0$ = ... cm

2) $2l_{red2} = \frac{J_0 R}{1{,}3\,\Theta_W}$ = ... "

3) $l_{red3} = \alpha\,\frac{J_0}{J_{PKZ}}$ = ... "

4) $2l_{red4} = \frac{J_0 a^3}{31{,}2 J_{axKZ} R k}$ = ... "

5) $2l_{red5} = \frac{1{,}2 J_0 a}{F_{FZ} R k}$ = ... "

6) $2l_{red6} = \frac{J_0 R^2}{3{,}9\,\Theta_{wk}}$ = ... "

7) $2l_{red7} = \frac{2{,}4 J_0}{F_{wk}}$ = ... "

8) $2l_{red8} = 1{,}8\,\frac{b^2 + h^2}{b^3 h^3}\,(a + h)^2 (R - 0{,}5\,d)\,\frac{J_0}{Rk}$ = ... "

Sum 1 . . . 8 = ... cm

9) $l_{red9} = \frac{J_0 a R}{J_{PKZ} k^2}$ = ... "

Sum 1 . . . 8 minus 9 = ... cm
= reduced length of crank portion of shaft.

the crank or in determining the characteristic rates of oscillation, although k is used in the former of these two operations. In Table 3 k appears in members 4 to 9. In this case the reduced lengths of the individual parts resulting from torsion, flexure and displacement of crank pins and cheeks, are to be computed. Members 1 to 8 are to be used with a positive sign, member 9 with a negative one. The sum gives the reduced length of the cranked portion sought. The numerical calculation is so simple that it can be carried out even by slightly trained assistants.

The torsional elastic length of the crank cheeks (11) in Table 1 and No. 8 in Table 3) is represented by R—O, $5d$, a value which showed a remarkable agreement between the measured and computed torsion in the numerous experiments carried out by the A. E. G.

Similar formulae can also be constructed for determining the reduced length of crank shafts of engines with pistons running in opposite directions, although the deductions are considerably more complicated owing to the triple cranking for each cylinder and the number of calculations that have to be made is considerably greater. In this connection also a very gratifying agreement was found between the calculation and the actual experiment. The calculations are not reproduced here, however, as this type of construction is now but little used in Germany. This is quite contrary to the practice in England and America, where this system is again receiving increased attention.—"Zeitschrift des Vereines deutscher Ingenieure," January 15, 1921, page 67.

OBITUARY.

GEORGE F. KUTZ.

Chief Engineer George F. Kutz died at San Francisco on the 11th of August, 1921, at the ripe old age of 84, from a general breaking down. He was born in Wilkesbarre, Pennsylvania, in 1837, and, after his schooling, he learned the machinists trade, with a view to becoming an engineer in the Navy. He took a post-graduate course in the machine shop at the Washington Navy Yard, and worked on the engines of the steam frigate *Minnesota* in 1855.

He entered the Engineer Corps in June, 1856, in a class of ten, standing No. 2 in that class, and was not quite 20 years of age. His first order was to the frigate *Niagara*, which vessel, with the British frigate *Agamemnon*, laid the first Atlantic cable. The Paying out gear was the invention, and design, of Chief Engineer Everett of the *Niagara*. The vessel was so low in the water, from the great weight of the cable, that the British press expressed fear of her safety. Bad weather made the voyage a hazardous one.

Mr. Kutz was sent to the *Atlanta* from the *Niagara*, in which vessel he went in the famous Paraguay expedition, an expedition hurriedly organized, and composed of small purchased vessels of questionable sea-going qualities. Mr. Kutz acted as chief engineer of the *Atlanta*, and though so young he was one of the most resourceful engineers in the little fleet, and was ever self-reliant. He was promoted from a third assistant to a first assistant engineer (which was unusual) and sent to the *Saginaw*, a very small side-wheel vessel being built at San Francisco for use on the China and India station, where she could ascend the rivers in safety.

At the breaking out of the Civil War, in 1861, the *Saginaw* was ordered home, when Mr. Kutz was promoted to Chief Engineer and ordered to the *Monongahela*, a large, high-powered steam sloop, which vessel was in the drastic fighting at New Orleans, Baton Rouge, Grand Gulf, Mobile, etc.

After the war he served in the *Ticonderoga* in the Mediterranean, then in the flagship *Pensacola*, and in the *Benicia* in the Pacific, and later in the flagship *Tennessee*. He was a long time Chief Engineer at the Mare Island Navy Yard, and he served on the Board of Examiners at Philadelphia for several years, and retired, voluntarily, in 1896, with the rank of rear admiral.

It was the writer's fortune to sail with him in the flagship *Pensacola* in 1872-3, in the Pacific, at a period when much of the cruising was done under sail, keeping the sea for many days, which is wearing on the best of us. It was remarkable how bright and cheerful Mr. Kutz was, when others were weary and peevish; his bright, witty remarks, good nature, and happy disposition made him the life of the ward room mess. One day the fleet surgeon was lamenting over the long voyage, for the vessel had then been 37 days close hauled on one tack, in the trades, without any change in the sails (except now and then to taughten a brace or a halliard), and Mr. Kutz turned with his happy face and said: "Why, Doctor, there are people at Cape May at this very time paying $15.00 a day for less pleasure than this."

Though by no means a large man, he was spunky and pugnacious, with a keen sense of right, never aggressive, but quick to defend. He had many words at easy command, and never failed to use exactly the correct word to express his thought. He detested a mean man or a mean act; he was as free from jealousy (that faculty that makes nearly all the trouble) as any man the writer ever knew. He was as particular to commend a proper act as he was to condemn a fault.

He was an indulgent husband, a generous and affectionate father. He is survived by three sons, one of whom is a pay officers and another a lieutenant in the service; and by one daughter, a remarkably beautiful woman, who is the wife of a retired officer in the Marine Corps.

For several years we have looked up to Kutz as the last of the "old chiefs," *i. e.*, the men who were chief engineers during the Civil War.

"One by one the roses fade."

G. W. B.

BOOK REVIEW.

A MANUAL OF MARINE ENGINEERING, A. E. SEATON. (D. Van Nostrand Co.) The high character of this book has long been known to the Marine Engineering world and its popularity has been attested by the number of editions that have been required. But during the past ten years, marine engineering has undergone some remarkable changes and it is not believed that these are adequately covered by the last edition. In its present form, the book should be regarded more as a history of past performances than as a description of present practice.

ASSOCIATION NOTES.

The regular meeting of the Society was held at Washington, D. C., on Tuesday, October 4, 1921. The following were nominated for officers for the year 1922:

For *President:*

Rear Admiral C. W. Dyson, U. S. N.

For *Secretary-Treasurer:*

Commander S. M. Robinson, U. S. N.
Commander F. J. Cleary, U. S. N.

For *Member of Council:*

Rear Admiral B. C. Bryan, U. S. N.
Captain Robert Stocker (CC), U. S. N.
Captain F. L. Pinney, U. S. N.
Captain E. C. Kalbfus, U. S. N.
Captain P. B. Dungan, U. S. N.
Commander S. C. Hooper, U. S. N.
Commander R. A. Spruance, U. S. N.
Commander H. S. Howard (CC), U. S. N.
Lt. Commander M. R. Daniels, U. S. C. G.
Lieutenant H. F. Johnson, U. S. C. G.

Commander F. J. Cleary declined to allow his name to stand for Secretary-Treasurer.

The President was authorized to appoint a committee to consider and present to the membership of the Society a revision of the By-Laws.

A vote of appreciation was tendered Commander J. S. Evans, U. S. N., for his splendid work as Secretary-Treasurer of the Society.

It was voted to place before the membership the matter of holding a banquet during 1922. Provision for voting on this proposition was made on the ballot for officers.

Votes for officers will be counted at a meeting of the Society to be held at the Navy Department on Friday, December 30, 1921.

At a meeting of the Council of the Society held on October 4, 1921, Commander J. S. Evans, U. S. N., tendered his resignation as Secretary-Treasurer, due to his detachment from duty in Washington. The Council appointed Commander S. M. Robinson, U. S. N., Secretary-Treasurer for the unexpired term of Commander Evans.

The following new members and associates have joined the Society since the publication of the last JOURNAL:

MEMBERS.

Alford, T. N., Lieutenant Commander, U. S. N.
Brandt, B. F., Ensign, U. S. N.
Connolly, Joseph A., Ensign, U. S. N.
Cullen, J. J., Lieutenant, U. S. N.
Curtin, L. W., Ensign, U. S. N.
Dessez, J. H. S., Lieutenant Commander, U. S. N.
Lake, Burton G., Ensign, U. S. N.
Leighton, Bruce G., Lieutenant Commander, U. S. N.
Livingstone, W. G., Ensign, U. S. N.
Lynch, John F., Ensign, U. S. N.
McNamar, J. A., Ensign, U. S. N.
Mackay, W. E., Lieutenant, U. S. N.
Merrill, A. S., Lieutenant Commander, U. S. N.
Murphy, E. J. A., Lieutenant, U. S. N.
Nelson, A. D., Ensign, U. S. N.
Perlman, Benjamin, Lieutenant Commander, U. S. N.
Pickhardt, A. V. S., Lieutenant Commander, U. S. N.

Porter, William H., Lieutenant Commander, U. S. N.

Roberts, J. A., Jr., Ensign, U. S. N.

Smith, C. E., Ensign, U. S. N.

ASSOCIATES.

Boyd, William, Managing Director, Workman, Clark & Co., Ltd., Belfast, Ireland.

Gardner, Thomas, Chile Exploration Co., Tocopilla, Chile.

Hehre, F. W., Professor, Electrical Engineering Department, Columbia University, New York City.

Taylor, H. B., Chief Turbine Engineer, U. S. Shipping Board, 45 Broadway, New York City.

The Secretary again brings to the attention of every member the necessity of keeping him advised as to all CHANGES IN ADDRESSES. In no other way may you be assured of receiving your JOURNAL promptly.